POWER

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Profit Repellers

Admitting that more people try to get orders these days than to give them, it still pays the host to be hospitable to representatives of legitimate business.

On a recent occasion a misinterpretation of the purpose of a man's call, because of what was on his business card, lost a manufacturer an order running well into six figures.

"Another one of those salesmen! Tell him I can't see him!"—The caller urged the executive's secretary to repeat his request for an interview a second and third time. On the *last* effort, the busy executive said he guessed he could determine the disposition of his own time and when he said something he meant it—and that was that—and it was final! So the unrequited caller was actually forced to apply to someone for the name of another builder of the sort of equipment he wanted to purchase.

Now, that same busy executive would be among the first to resent any such treatment of his salesmen as he had given to the man who, in this case, wanted not to take an order but to give one.

Men not otherwise inclined to simple business courtesy should realize that their desire for isolation may be effectively driving profits elsewhere.

EDITORIALS

How to Keep Poor

"Money begets money" is a maxim old and generally true. A new revision, particularly applicable to this year of 1932, might be "Poverty begets poverty."

It is always so with the poor. The man without cash in pocket feels that he cannot afford the moderate cost of preventive medicines and as a result may pay three times as much to save life in emergencies.

It is the same with business and industrial concerns, many of which find themselves today for the first time among the poor. Need for cash has, of course, resulted in many real economies, but it has also led to new waste in many cases due to actions that parallel rather closely those of the "poor" individual.

Almost every industrial power plant, for example, shows possibilities of certain investments—small or large—that would yield from 25 to 200 per cent yearly return on the investment by the elimination of fuel waste and other inefficiencies. To pass up these opportunities, particularly those that would pay out in a year or two, means a depletion of assets that is hardly short of criminal if it can in any way be avoided.

It will certainly pay every plant, however severe its financial stringency, to survey its own wastes, list the investments called for and estimate the net return on each. It should be possible to take care of some of the smaller items with highest returns by cash investment. Larger investments in engines, stokers, turbines, etc., yielding say 25 per cent or more, may apparently present a more difficult problem. But here it is sometimes easy to arrange for payment out of savings. In such cases, the last excuse for postponement is removed.

The problem of the poor, today as always, is how not to get poorer. Here is a practical solution as far as many of the industrial power plants are concerned.

Oversize Motors on Undersize Feeders

Under-size lines supplying over-size motors are an undesirable combination too frequently found in power applications. There always has been a general tendency to select line conductors too small and motors too large, notwithstanding that such a practice is expensive and may lead to serious operating difficulties. In many cases where feeders are under-sized, a proper distribution of electrical equipment costs in the original installation would have resulted in a satisfactory combination of motors and power lines at no added expense.

Over-size induction motors are undesirable because of their low power factor even when feeder capacity is ample for the load. Where lines are too small, overmotored loads tend to aggravate an already bad condition. The low-power-factor load current may cause a voltage drop that will seriously affect operation of motors and control. The torque of an induction motor varies as the square of the voltage applied to its terminals. Because of this, cases are on record where line losses have reduced the voltage so that the motors could not start, or so that they have stalled during peak loads. Operating difficulties and delays are only part of the penalty that must be paid for under-size lines. A ten per cent excess in line loss is equivalent to an equal loss in motors or power plant efficiency and represents an equal increase in operating cost.

Excess capacity put into lines compensates, at least in part, for the added investment by reducing power losses, improving equipment operation and in ability to supply additional load. Comparable advantages can seldom be claimed for overmotoring. This should be kept in mind when making power applications.

Measurement of Turbine Input

When all steam turbines were simple condensing units, determination of the steam flow was not difficult, as the condensate could be metered. The addition of bleeder heaters and steam reheaters, the interposing of deaerators, generator air coolers, oil coolers, steam-jet air pumps and evaporators has made impossible the former simplicity of measurement. While one can still meter the total steam entering the turbine with substantially the same accuracy as formerly, this may not mean much, since all of it does not pass completely through the turbine. In any case the total steam to a turbine operating on regenerative or reheating cycles means little. What it is required to determine is the heat rate of the turbine, that is the difference between the heat supplied at the throttle and at the reheater and the heat returned in the condensate.

At present the heat rate determination requires the measurement of numerous quantities and involves much labor. Have inventors ever considered the possibilities of a "heat meter" with which the total heat supplied can be integrated so that the total heat in the feedwater can then be deducted. The problem is complicated, but such a meter, satisfactorily developed, will find wide applicaion.

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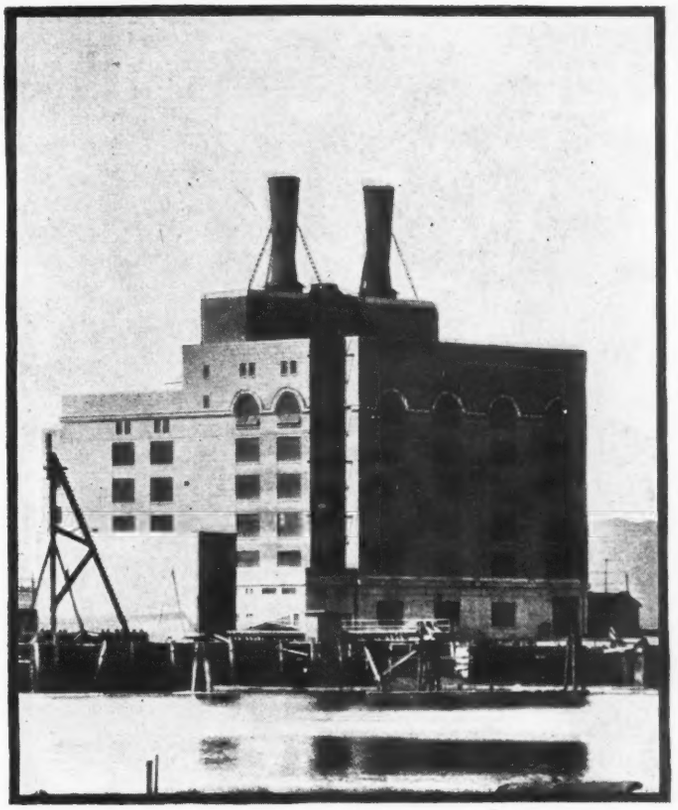
POWER Stands for . .

1. Making Power When It Should Be Made
2. Buying Power When It Should Be Bought
3. Cheaper Power Through Modern Equipment
4. Easier Financing of Equipment Purchases
5. Better Use of Byproduct Heat and Power
6. Operating Methods That Save Money
7. Less Waste in Transmission and Application
8. Prevention of Smoke, Within Reason



City of Tacoma's new steam plant as viewed from Hylebos Waterway

Another steam plant in the heart of the hydro-power country, this $2,000,000 first section of a 100,000-kw. station was first brought to full load on May 11

Tacoma's New Steam Power Plant

is Auxiliary to Municipal Hydro System

By LLEWELLYN EVANS, F. T. DOWNING and A. F. DARLAND

Engineers for City of Tacoma, Washington

COAL mined in Washington State is getting a new market with the completion of the 25,000-kw. steam plant of the City of Tacoma, Wash. The decision to burn powered coal in the plant was reached after the City had received bids on five-year supplies of fuel oil and of coal mined within the state. The new plant, costing $2,000,000, is the first unit of a 100,000-kw. station and will serve as an auxiliary to the three hydro stations now supplying Tacoma's system.

The plant is located at East 11th St. and Hylebos Waterway in the tideflats industrial area, on a 20-acre tract of land. The site is connected by rail through the City Belt Line with four trans-continental railways and by water with local boats and ocean-going ships, assuring transportation for fuel. Cooling water is available from Hylebos Waterway for the first unit and for later units canals are to be extended to connect with Wapato Waterway. This will provide an unlimited supply of salt water at temperatures below 55 deg.

Plant buildings are of reinforced concrete tile and brick construction, except for a portion of the west wall. This is of steel panel construction to provide for future extension of the plant. The plant building proper is 92x163 ft. exclusive of intake structure.

Storage is provided at the plant site for 8,000 tons of coal for emergency consumption. Storing and reclaiming of coal is done by locomotive crane.

Electrical Plant

The generator is contructed for an armature temperature rise at full load of 60 deg. C maximum, and has direct-connected main and pilot exciters. Air coolers of the fin-tube type using sea water are supplied for normal operation. A battery of carbon dioxide cylinders is installed for protection in case of fire in the generator. The generator is arranged for operation as a synchronous condenser when disconnected from the turbine.

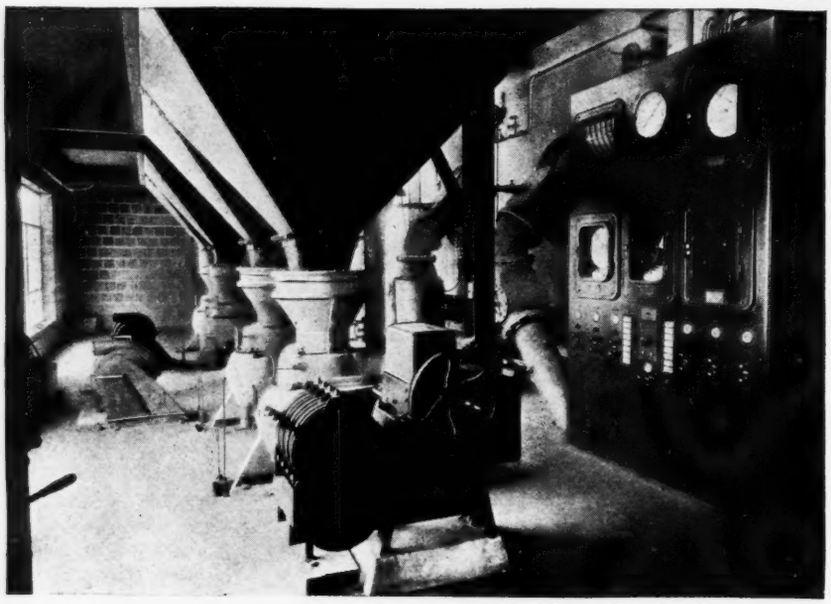
Electrical energy is delivered directly to the City's distribution system at 13,800 volts, 3 phase, 60 cycles. A bank of 5 per cent reactors is used to augment the inherent reactance of the generator.

Power for station service is supplied from the station through a 13,800 to 2,200-volt transformer bank of 3,000-kva. capacity and from the city system by an open

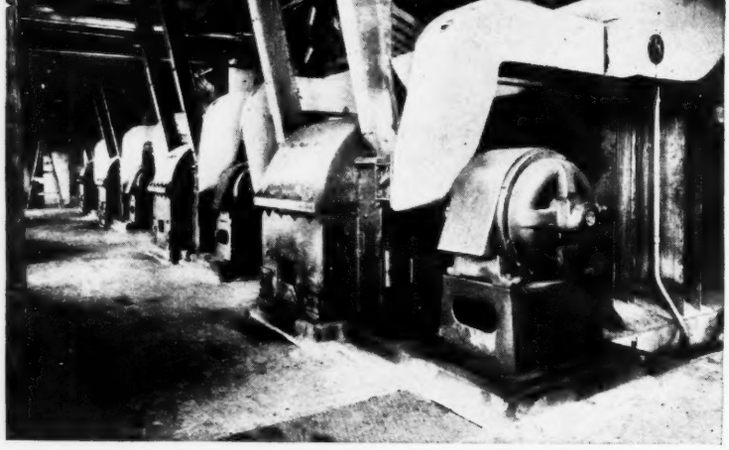
delta bank consisting of two 1,000-kva. transformers of similar voltage rating.

The main station service bus, consisting of three sections, is equipped with truck-type switch gear. One section is fed from each of the main service transformer banks, while the third is identified as the house service bus. Distribution buses at 2,200 volts are placed in both the boiler and turbine rooms and supplied with energy from the main station service bus by means of duplicate feeders and switches that automatically select the energized feeder. Station service motors of 50 hp. and over operate at 2,200 volts, and smaller motors operate at 440 volts, except for special instances where direct current is required. All alternating current motors are the constant-speed type started at full line voltage.

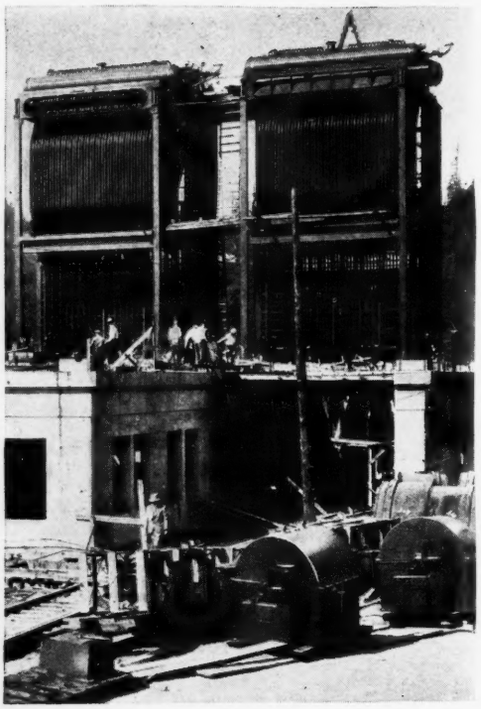
A motor-driven spare and pilot exciter provides breakdown service for the main generator and coal feeders and also supplies direct current energy for the crane.



Boiler room floor showing four mill feeders in front of the boiler and the combustion control in front of the instrument panel



Two pulverizers per boiler are required to supply coal to generate 184,000 lb. of steam per hour



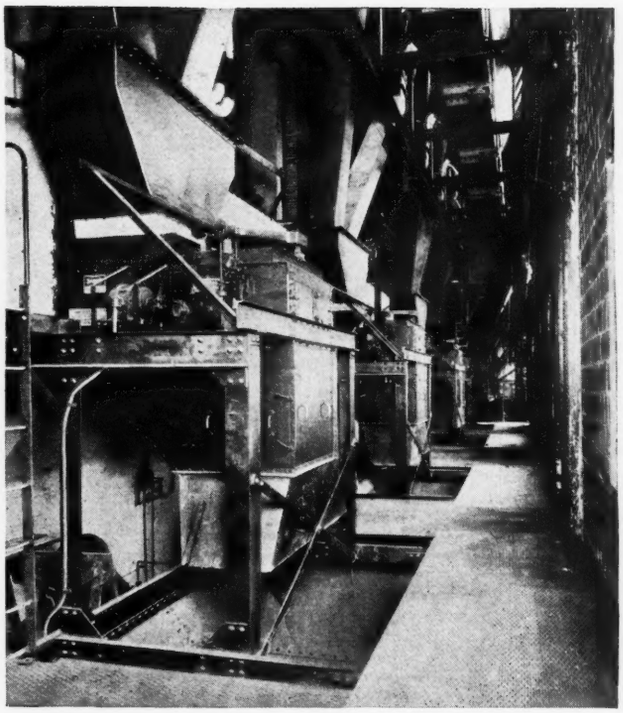
The two boilers were erected before the building that houses them

A control room overlooks the turbine room and the transformer and switchyard site. The usual metering and controls are placed on a benchboard, and the principal relays are located on a vertical board to the rear. A voltage regulator and battery board complete the present switchboard equipment. The quick response type of excitation is employed.

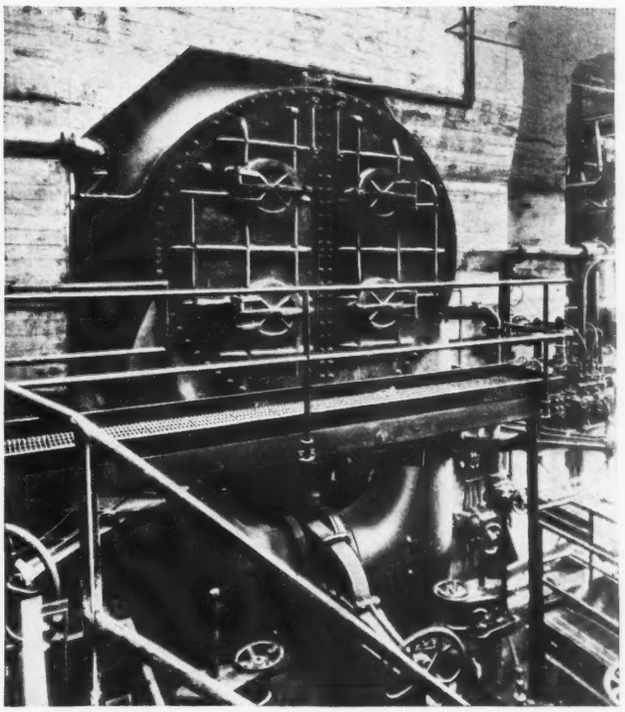
In the initial installation one turbine generator with a nominal rating of 25,000 kw. and a maximum rating of 29,000 kw. for 4 hr. has been installed. The single-cylinder turbine has fourteen stages, and operates at 385-lb. steam pressure, 725 deg. temperature, and at a speed of 1,800 r.p.m. It is bled at the third, seventh and eleventh stages for heating of feedwater with a steam rate at full load of 9.4 lb.

The single-pass condenser has condensing surface of 18,000 sq.ft. exclusive of the after-cooler. Condenser tubes of $\frac{3}{4}$-in. diameter are rolled into the tube sheets at both ends, and a rubber expansion joint is provided between water boxes and the shell to allow for longitudinal expansion. As the shell is partially hung from the turbine exhaust flange, that portion of the weight of the shell and water boxes not taken care of by this method is supported by shoes on adjustable springs.

Condenser circulating water is supplied by two 24-in., 16,000-g.p.m. pumps, and condensate is removed by



A separate weigh scale is provided for each pulverizer giving a record of each mill's performance



Consenser auxiliaries are located along the wall to the right of the picture

two 12-in., 750-g.p.m. 2-stage pumps. Performance of the unit is based on 29-in. vacuum at 25,000 kw. load using 34,000 g.p.m. of 59-deg. cooling water.

Heat Cycle

The condensate pumps discharge through the steam jet air pump condenser and the eleventh stage or low-pressure closed bleeder heater to a deaerator. The deaerator receives the various plant drips, make-up water from an evaporator exhaust from a turbine-driven boiler feed pump and steam from the seventh stage bleed point, regulated by an automatic bleeder-flow control valve. The deaerator is provided with an integral storage and surge tank to maintain suction pressure on the feed pumps. The third stage bleeder heater is provided with a drainer which discharges into the deaerator.

When starting up after a prolonged shutdown of plant, an auxiliary condenser is provided to condense vapor from the evaporator supplying make-up to boilers.

The boiler feed pumps; one 750-g.p.m. turbine driven, one 750-g.p.m. and one 250-g.p.m. motor driven, receive their supply of water from the deaerating heater, and at light loads from this heater and a distilled water storage tank located under the heater. One boiler feed pump of 750-g.p.m. capacity is sufficient to supply the total feed-water demand of both boilers.

Two 1,650-sq.ft. boilers are installed, each capable of providing sufficient steam to carry two-thirds of the rated turbine generator capacity. These boilers each generate normally 150,000 lb. of steam per hr. at 420 lb. gage pressure and 750 deg. They are of the bent-tube, drum type with integral convection superheaters containing 4,300 sq.ft. of heating surface. The furnace has air-cooled front and side walls, a water-cooled ash hopper and a bare-tube, back-wall water screen. The water screen is fed by water from the lower drums through multiple downcomers and headers to insure constant circulation at all ratings. The effective furnace volume is 14,000 cu.ft., and the B.t.u. release about 18,000 B.t.u. per cu.ft. maximum.

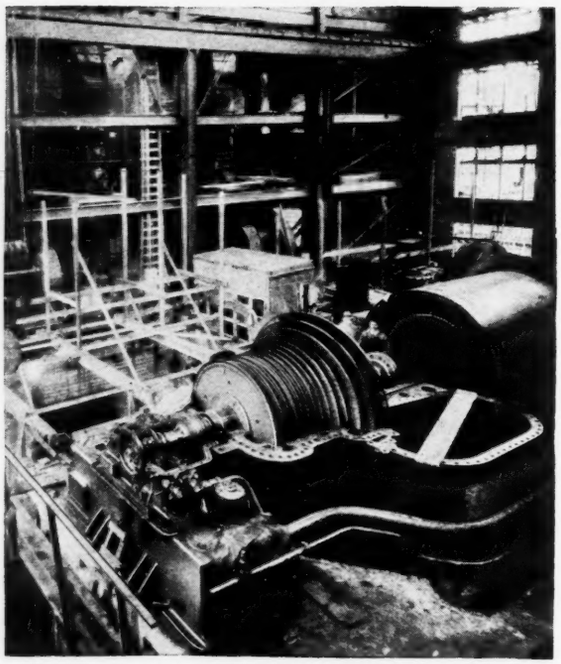
Reinforced plastic cross baffling is used to reduce friction drop and increase heat absorption. The baffles are arranged to choke gases in the last pass to maintain constant velocity through boiler passes.

Each boiler is equipped with one forced draft and one induced draft, multivane, damper-controlled fan of 65,000 c.f.m. and 105,000 c.f.m. respectively. Both fans are driven by two 2,300-volt squirrel-cage motors, one high- and one low-speed direct connected.

The coal pulverizing equipment per boiler consists of two Atrita duplex pulverizers, driven by 2,300-volt, a.c., enclosed, 125-hp. motors which supply fuel to two flare type burners. Four feeders, two for each pulverizer, are operated through an intermediate shaft, driven by two d.c. 250-volt motors using silent chain drive. The burners are constructed so as to burn gas or oil if expedient or found necessary.

The ash handling system used is an integral part of the boilers. Water cooled ash hoppers are provided to collect the refuse. From these it is sluiced to ash tunnels by oscillating high-pressure nozzles located in the hopper area. From these tunnels the ash is sluiced by hydro-jet nozzles to an adjacent ash sump. A high-pressure specially designed motor-driven ash pump, constructed to handle ash-laden water, pumps this highly abrasive material to an outside ash dump.

Preheaters provided for these boilers are of the regenerative, vertical, rotor type, motor driven. It is expected that with these preheaters the boilers will operate at an over-all efficiency of 85 per cent. Coal delivered by rail is dumped into a 50-ton track hopper from which it is removed by a plate feeder and discharged to one of two 60-ton per hour elevators. A screw conveyor of the same capacity distributes the coal to a 400-ton steel bunker which is sufficient fuel for 14 hours operation at full load. A coal crusher is located so that it can be



Construction view of the turbine room showing boiler room in the background

fed from the plate feeder by a 75-ton per hour belt conveyor when the size of coal used exceeds 1½-in. screen.

Coal from the bunker is fed through spouts by gravity to automatic weigh scales, which feed the weighed coal to a battery of 5-ton hoppers. From these hoppers the coal feeds by gravity to the feeder hoppers of the coal pulverizers.

Combustion control is provided so that the two boilers can be operated as a battery or as separate units depending upon the conditions of load. The control functions to regulate the amount of fuel and air for changing load on each boiler independently or on both boilers simultaneously, automatically or by pushbutton. It consists mainly of motor-operated control drives for fuel feeders and forced and induced draft fans of each boiler; regulating the speed and damper openings of each.

The high-pressure steam and water piping is all small tongue and groove facing with A.S.T.M. standard flanges. Motor-operated line protection valves are provided in all high-pressure steam lines from the main steam header.

Steam conditions of 390 lb. gage and 725 deg. at the turbine throttle were decided on for three reasons, namely: 1. The turbine cost was no more for 400 lb. than it would have been for a 250-lb. unit. The costs per kw. above 400 lb. would materially increase. 2. Extraction feed water heating was expedient by virtue of the high steam pressure and temperature at three bleeder taps. 3. As the mechanical and thermal efficiency of a turbine increases with the total steam temperature, the high reasonable temperature of 725 deg. was decided upon.

The design and construction of the plant was started with J. L. Stannard as chief engineer; E. B. Miller, supt. of steam design; F. T. Downing, mechanical engineer, Alvin F. Darland, supt. of electrical design and construction; Leon R. Evans, electrical engineer; J. V. Gongwer, supt. of hydraulic design and construction, and V. R. Rathbun as resident engineer. After the death of J. L. Stannard and the resignation of E. B. Miller the remaining engineers completed the work.

Suction Lifts For Pumps

By DONALD McFEE

THE first problem confronting an engineer who finds that he must install a pump is that of suction lift. Assuming that the pump is of the centrifugal type, how much of a lift can be expected of the unit? Introducing another factor, that of water temperature, how much below the pump can the water level be in the suction tank if the water is at, for example, 130 deg. F.?

If the engineer is one of those who accumulate engineering data, he will reach into his book shelf for a pump catalog, for almost all of them include a chart showing the suction conditions for any given water temperature. One is shown in Fig. 1.

Examination of the chart reveals that the pump manufacturer recommends for water at 130 deg. F. a suction lift (the distance the water level in the suction well is below the pump) of not over 6.5 ft., as indicated by Curve *a*. The chart also includes the curve *x*, which is stated to be the theoretical head obtainable with different temperatures, and the curve *m*, termed the maximum results obtainable under the most favorable conditions.

Engineers who do not care to use charts blindly may wonder where and when this chart originated. It seems that it first appeared in an engineering magazine about 25 years ago. This age should not be held against the chart save on the grounds that it is based on erroneous thinking and does not represent the suction conditions necessary to obtain satisfactory operation.

Assuming for the time being that the curve representing the theoretical suction is correct, the first thing that attracts one is that the difference between curves *x* and *m* is only 7.5 ft. at 60 deg. F., but at 150 deg. F. the difference is 19 ft. To bring out this point, examine Fig. 2.

The chart, Fig. 2, has been constructed by transferring curve *x* from Fig. 1 and setting down the difference between curves *x* and *m* in Fig. 1. It will be seen that the increment curve *x*—*m* includes a decided hump that has no excuse for existence. If the *x*—*m* value at 60 deg. F. should be 7.5 ft., then the value of 19 ft. at 150 deg. is without rhyme or reason. There is every ground for stating that the increment should be uniform throughout the temperature range.

Furthermore, the theoretical curve *x* in Fig. 1 is not based on correct premises. In its construction the value of the vapor pressure at each temperature point has been subtracted from the standard barometric reading of 34 ft. to give the pressure forcing the water up the suction pipe. In the construction of a curve of this sort the variation in water density must be included. For example, the standard atmospheric pressure is 14.68 lb. per sq. in.; water at 150 deg. F. has a vapor pressure of 3.71 lb.; the difference between these two values, namely, 10.97 lb., forces the water up the pipe. Since 150-deg. water has a density of 61.2 lb. per cu.ft., a head of 2.35 ft. is equivalent to 1 lb. per sq.in. pressure. Obviously the 10.97 lb. of net pressure acting on the surface of the suction well will support a column of water 10.97x2.35= 25.78 ft. high. This, then, is the real maximum theoretical lift.

If this process be carried out for various points it will be found that the curve obtained is that at *x* in Fig. 3.

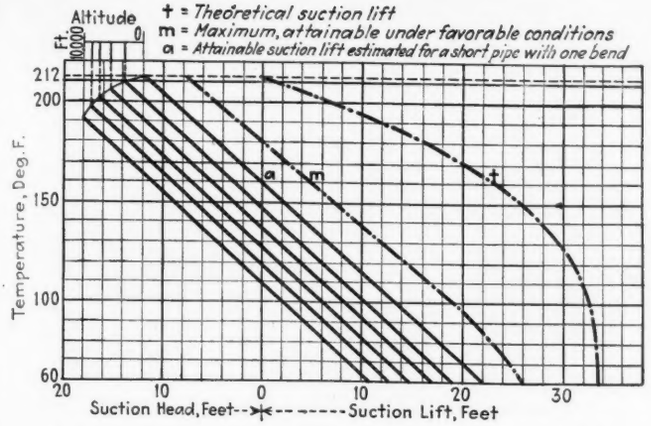


Fig. 1—Chart frequently used for determining suction lift with centrifugal pumps

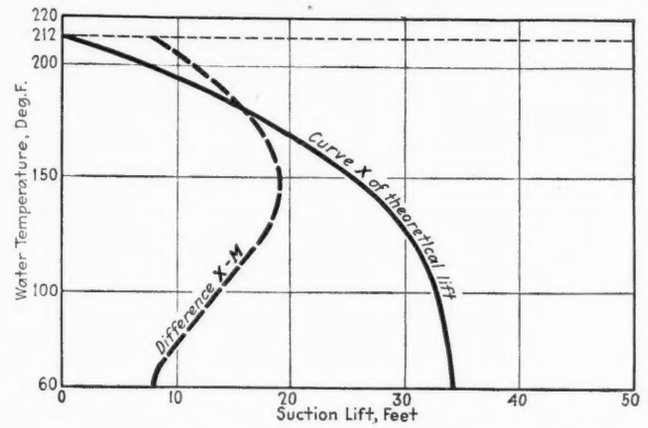


Fig. 2—Chart showing difference between theoretical lift and maxamum lift given by curves *M* of Fig. 1

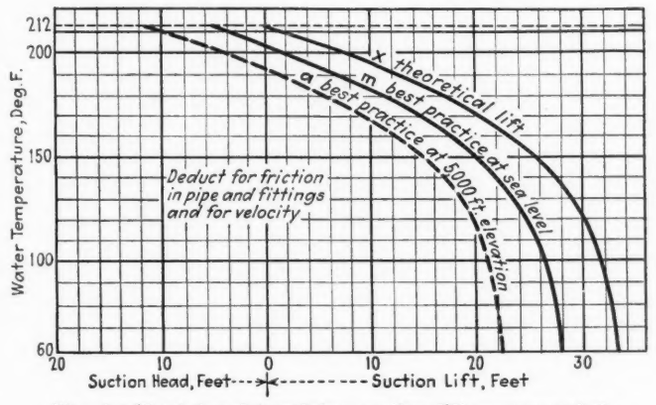


Fig. 3—Chart for determining suction lift recommended by author

As for the best lift in practical operation, curve *m* in Fig. 1 is in error. To prevent cavitation at the entrance to the impeller of a centrifugal pump it is necesssary to have the water reach the pump under a slight positive head. Experience indicates that 5 ft. is ample.

To construct a curve showing this "best operating head" it is only necessary to deduct 5 ft. from the values of curve *x*, giving curve *m* in Fig. 3.

The same process may be employed to find the best head at any altitude. In Fig. 3 this has been done for a 5,000-ft. altitude. In the calculations the pressure at this altitude is used instead of 14.68 lb.

In this discussion nothing has been said about velocity head and friction head. Pumps generally are provided with suction-pipe areas of a size to hold the velocity to below 10 ft. per second. The velocity head is then less than 2 ft., so it may be ignored in most cases. The friction head is another and sometimes serious matter; it must be considered when one is solving a suction problem.

It is impossible to include friction head in a curve without definite information of pipe diameter and length. An engineer should decide upon the probable length of pipe and upon its diameter. With this data use may be made of the Hazen & Williams formula for friction. This is:

$$V = 1.318\ C\ R^{0.63} S^{0.54}$$

where

V = Velocity in feet per second.

R = Cross-section area of pipe ÷ inside circumference, $= \frac{\pi}{D}$, in feet

S = friction head ÷ pipe length.

C = Coefficient of pipe roughness, varying from pipe 100 to 150 according to velocity and pipe diameter.

A quicker method is to use the pressure flow chart in Mark's Handbook.

For example, let a 6-in. pump be assumed to be handling 880 gal. of water per minute. This is equal to 1.96 cu.ft. per second making the velocity about 10 ft. per second. The chart in Mark's shows that the friction loss in 1,000 ft. of 6-in. pipe would be 70 ft. A pump suction line of, say 50 ft., including a horizontal run, would entail a loss of head due to friction of approximately 3.5 ft. Consequently the suction lift as found from Fig. 3 must be reduced by 3.5 ft. As the velocity is 10 ft. per second, the velocity head will be $\sqrt{\frac{V^2}{2g}} = \sqrt{\frac{100}{64.4}}$ $= 1.25$ ft. The total loss is then $3.5 + 1.25 = 4.75$ ft. Then again, elbows and valves entail a loss in head; the deductions to be made may be found in any handbook.

The suggestion is made that the curve in Fig. 3 be employed in finding pump suction lifts. Common experience has proved to many that Fig. 1 gives too low a lift and if followed entails unnecessary pipe work and excavation. From the lifts obtained from Fig. 3, deduction must be made for pipe and fitting friction losses and for velocity head.

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Steam Pipe Insulation Underground

WHEN steam pipes are laid underground, the insulating capacity of the earth reduces the economical thickness of heat insulation. According to Dr. Ing. I. S. Cammerer the insulation thickness as compared with a pipe in air can be reduced one-half, and the heat loss from underground uninsulated pipe should not be assumed greater than double the value of the heat loss of insulated pipes. The temperature of the ground around a pipe carrying steam at 350 deg. is influenced for a distance of 16.4 ft. The temperature around the pipe is distributed, not in a circle, but in an ellipse with the larger axis horizontal and the smaller one displaced eccentrically downward. Tests indicated a conductivity of loamy earth of 4 to 4.8 B.t.u. per sq.ft. per in. per deg. F. for the zone around the pipe.

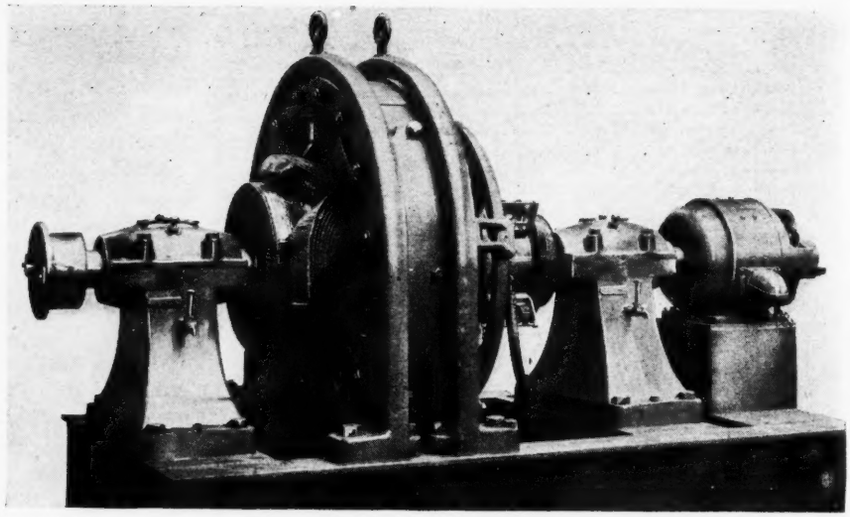

Fig. 1—Compensated-type 60-kw. 125-volt main exciter with a 250-volt pilot exciter

By IRVING C. SMITH

Electrical Engineer, Elliott Company

Compensated Exciters Designed for Quick Response

Response to voltage changes of the non-compensated type of exciter is compared with that of the compensated type and the factors that influence the rate of response of an exciter are discussed. A 60-kw. compensated-type exciter with pilot exciter having a rate of response of 3,380 volts per second is described

ADVANTAGES of quick-response excitation in maintaining system stability and steady voltage regulation have been recognized for some time. One of the important factors in such an exciter system is the character of the source of excitation.

It is more economical to regulate the generator voltage by changing the exciter voltage than by cutting resistance in and out of the generator-field rheostat. Accordingly, it is customary for the voltage regulator to work on the exciter field rheostat to regulate generator voltage. If it is desired to realize the peculiar advantages obtainable from extremely rapid correction of generator excitation, it is necessary that the exciter voltage respond quickly to changes in exciter-rheostat resistance.

There is usually little to be done to make an alternating-current generator favorable to voltage response, inasmuch as the details of its design are fixed by other considerations. If a generator field is wound for 125-volt excitation, however, it will usually have a low inductance and be responsive enough to justify considerable attention being paid to obtaining good exciter response.

Consider the simple case of an alternating-current generator independent of any other system and excited by a self-excited exciter, Fig. 2. Suppose the generator is operating on no load at V_1 volts and F_1 amperes field current, Fig. 4. Then suppose that half load is suddenly thrown on and the field current is not changed. The voltage of the generator will drop to V_2 and stay there. If it is desired to raise the voltage to V_1, the field current will have to be changed from F_1 to F_2. If the resistance of the generator field is R_g, then the exciter voltage will have to be changed from F_1R_g volts to F_2R_g volts, and the exciter load will have to be changed from F_1 to F_2 amperes.

To change the exciter voltage from F_1R_g to F_2R_g requires a change in exciter field current from f_1 to f_2 amperes, Fig. 5. If a voltage regulator is used to maintain constant generator voltage, the drop in generator voltage below V_1 volts causes the voltage regulator to reduce the resistance of the exciter field rheostat. If the required correction is large, the voltage regulator may short circuit the entire variable resistance in the exciter rheostat.

The building up of current lags the impressed voltage in both the exciter and generator fields during the transient condition. Therefore, while the generator-field current is lagging behind the exciter voltage, the exciter voltage and field current have time to build up to values higher than F_2R_g and f_2. The maximum value of exciter field current would be between f_3 and f_4 amperes, following the dotted line up from f_1. The reason that the exciter voltage and field current do not increase along the constant-resistance load curve is that the exciter load current is something less than exciter volts divided by generator-field resistance, because of the lag of the generator-field current behind the exciter voltage.

While the generator voltage and field current are

approaching normal, the exciter voltage and field current drop along the field resistance line. If the exciter-field rheostat were to remain short circuited, equilibrium would finally be established at V_3 generator, volts F_3 generator-field amperes and F_3R_g exciter volts, f_3 exciter-field amperes. But as soon as the generator voltage has exceeded the value V_1, the voltage regulator inserts rheostat resistance into the exciter-field circuit, causing the exciter voltage and field current to drop along the dotted line to F_2R_g volts and f_2 amperes where equilibrium is re-established.

Consider now the case of a separately excited exciter supplying field-current to an independent alternating-current generator, Fig. 3. Assume the main exciter to have the same characteristics as shown in Fig. 5 for the self-excited exciter. When half load is suddenly thrown onto the generator, suppose the voltage regulator short circuits the main exciter rheostat. The exciter voltage builds up along a curve similar to the dotted curve shown in Fig. 5. The pilot exciter is flat compound wound and usually generates 250 volts. It is readily seen that the voltage applied to the separately excited exciter field coils is much higher than in the case of a self-excited exciter. This is particularly so when the generator field is wound for 125-volt excitation and the advantage of separate excitation in voltage response is apparent.

If the main exciter is non-compensated the field distortion due to armature reaction under load conditions causes high saturation under one side of each pole and reduces the total flux. An increase in field ampere-turns is therefore necessary to maintain the total flux at the required value. For this reason, the compensated exciter does not require as many field ampere-turns as the non-compensated exciter. Fig. 6 shows the field characteristics of compensated and non-compensated exciters of equal rating, so proportioned as to have the same no-load saturation curve when plotted between volts and ampere-turns per pole.

At rated load and voltage the non-compensated machine may require 25 per cent more ampere-turns than the compensated machine. Voltage regulators are limited in current-carrying capacity, the cost increasing with the capacity. On the basis of equal values of field current and flux in the two machines at rated load, the compensated machine will require only about 0.8 as many turns on its field as the non-compensated machine. The inductance is directly proportional to the number of turns on the field, providing the flux and field current are the same for the cases considered. Therefore, under the conditions assumed, the compensated exciter would have 0.8 as much inductance in its field as the non-compensated exciter. Considering the higher leakages in the non-compensated exciter, due to the greater number of ampere turns, this ratio would be still more in favor of the machine with compensating windings.

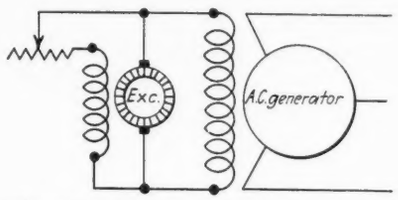

Fig. 2—Diagram of alternating-current generator and self-excited exciter

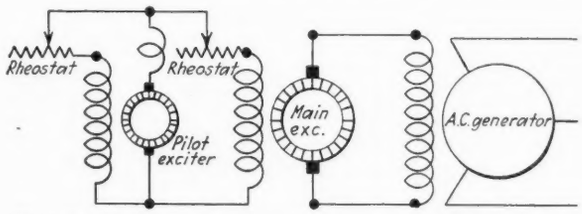

Fig. 3—Diagram of alternating-current generator and separately excited exciter

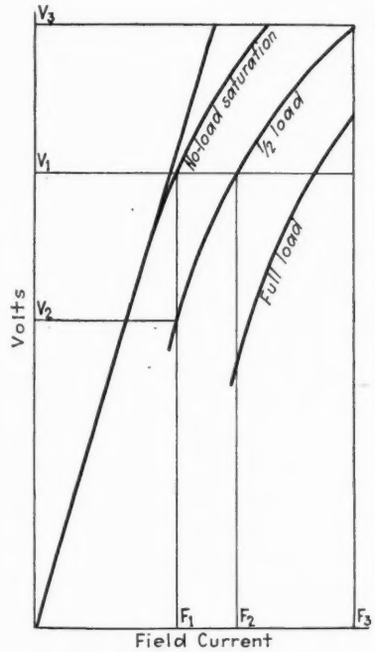

Fig. 4—Typical alternating-current generator characteristic curves, at no load, one-half load and full load

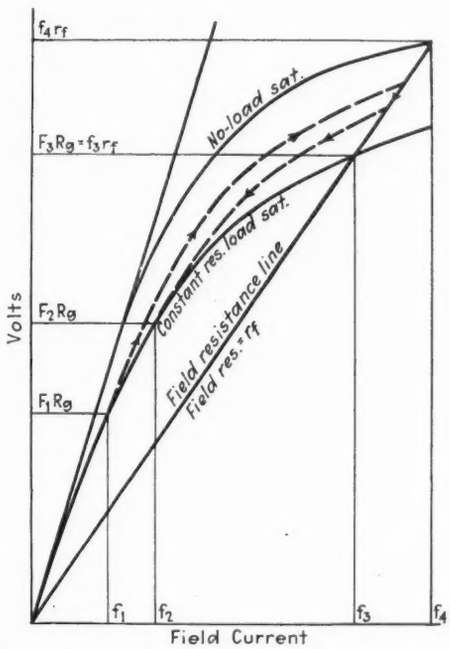

Fig. 5—Typical self-excited exciter characteristic curves, when connected to the generator field as a load

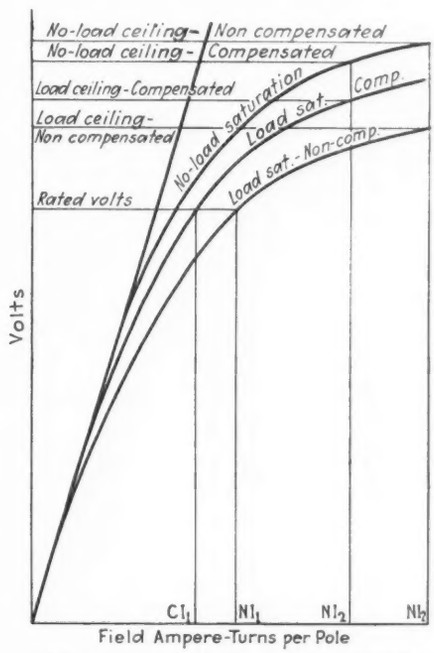

Fig. 6—Comparison of compensated and non-compensated separately excited exciters based on equal field currents

In Fig. 5 the constant-resistance-load saturation curve shows the relation between exciter volts and field current when connected to the generator field. The exciter load current increases directly at the exciter voltage except under transient conditions.

In Fig. 6 C equals the compensated exciters shunt-field-winding turns per pole; N equals non-compensated exciters shunt-field-winding turns per pole; I equals the field current in amperes at rated load and voltage; and I^1 equals the maximum field current.

The limiting current-carrying capacity of the voltage regulator is usually specified at the maximum excitation, that is when the field circuit has minimum resistance. If the current limit of the regulator is 20 amp., and if the pilot-exciter voltage is 250, then the minimum field-circuit resistance must never be less than $250 \div 20 =$ 12.5 ohms. If the field winding itself does not possess this much resistance, then a certain amount of fixed resistance is supplied in the rheostat. The voltage regulator will never short circuit the fixed resistance.

The ceiling voltage of an exciter is the maximum no-load voltage which corresponds with the minimum field-circuit resistance condition. The ceiling voltage of an exciter is significant because it is a measure of the potential force available to increase the generator-field current when the voltage regulator short circuits the variable-rheostat resistance. The exciter voltage actually effective in forcing the generator-field current to increase is some value between the no-load and load ceiling voltage. If the generator is operating at anything above half load, the effective exciter ceiling voltage is closer to the load ceiling than to the no-load ceiling. It is apparent from Fig. 6 that the compensated exciter supplies a higher effective ceiling voltage, since its load ceiling is higher than that of the non-compensated exciter.

There are several transformations of energy involved as the generator voltage is changed by varying exciter field resistance. The problem is to impress a source of electrical energy on the field of the exciter, transform this electrical energy into magnetic energy in the form of flux in the machine, then through the agency of the armature winding to transform this magnetic energy back into electrical energy available at the brushes of the exciter. In the process of each of these transformations, a certain amount of energy is lost. First, when the source of electrical energy is impressed on the exciter-field winding, there are such losses as stray or leakage fluxes, hysteresis and eddy-current losses in the iron and steel in the magnetic circuit. In the second transformation, as the armature current builds up, there are leakage fluxes set up which tend to oppose the building up of current and which set up eddy currents in the armature conductors. It is easily possible for these losses to diminish greatly the rate of response of a machine, therefore it is necessary that precautions be taken to minimize them.

The stray or leakage fluxes can be kept at low values by using short air gaps and low flux densities and by spacing the main poles well apart. Hysteresis and eddy-current losses in the magnetic circuit can be reduced by the use of laminated, varnished electrical sheet steel in the whole circuit, including poles and frame, and by keeping the flux densities low. Where rivets or through-bolts are used to hold these laminations together, the rivets and bolts should be insulated by means of insulating tubes and washers.

The leakage fluxes set up by the current in the armature winding can be effectively reduced by the use of compensating windings in the pole faces. Then, as the armature current changes in value the leakage flux has a correspondingly smaller retarding force. Another means of minimizing these leakage fluxes is the use of open, shallow, wide slots in the armature. A combination of short air gap and open, shallow, wide slots must be used with caution, since high pole-face iron losses, magnetic noise and tooth ripple in the voltage may become objectionable.

The eddy current loss in the armature conductors will not be noticeable, providing the depth of the conductors is kept within the limits of good design considered from the standpoint of armature heating.

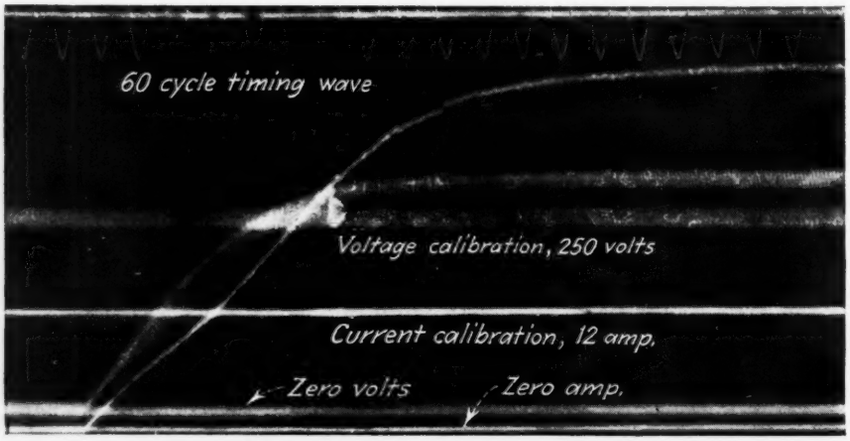

Fig. 7—Oscillogram showing building up of main-exciter voltage and field current when 250 volts from a pilot exciter was applied to its shunt field. The main exciter is compensated, shunt-wound and is rated at 60-kw., 125 volts, 1,750 r.p.m. This exciter was loaded with a resistance of 0.325 ohms when the oscillogram was taken

A shunt-wound exciter should always be used when speed of response is a consideration. A series field, consisting of a few turns of low-resistance copper, constitutes a rather effective damper winding and retards the changes in flux in the magnetic circuit of the exciter. If due consideration is given to these points in the design of the exciters, the best results will be obtained.

Recently, a quick-response exciter was built by the Elliott Company which had a rate of response of about 3,300 volts per second. The rating of this exciter is 60 kw., 125 volts, 1,750 r.p.m. and it is separately excited by a 250-volt pilot exciter, Fig. 1. The 60-kw. exciter is fully compensated, shunt wound, and built with laminated field, insulated bolts and rivets. Fig. 7 shows an oscillogram taken on this exciter when the main exciter armature was connected to a non-inductive resistance equal to that of the alternator field to be excited. With the exciter running at 1,750 r.p.m. and its field switch open, the field switch was closed, thereby impressing 250 pilot-exciter volts upon the main-exciter field and fixed-rheostat resistance such that the maximum field current would not exceed 36 amp. There was only one circuit in the main-exciter field.

The oscillogram shows the rate of exciter-voltage response to be an average of 3,380 volts per second from residual to 125 volts. A ceiling voltage under the above load conditions was 300 volts with a load current of 925 amp., or nearly twice rated current. Incidentally, it may be stated that the exciter commutated this current without any sparking whatever, even during the transient condition of rapidly increasing load current and voltage.

The oscillogram also shows the field current. It is seen that up to about 9 amp., the field current rises at a diminishing rate. The saturation curve of the exciter shows saturation to begin at 9 amp. field current. The inductance of the circuit is, therefore, substantially constant up to 9 amp. The only changing influence on the field current up to this point is the diminishing value of the voltage available to cause the current to change. In this exciter, it so happens that the saturation increases very rapidly between 8 and 10 amp., resulting in an appreciable change of inductance. This allows the field current to build up even faster above about 12 amp. than below, as shown by the change in shape of the field current between 10 and 12 amp. Of course, as the forcing voltage becomes more and more consumed by the *IR* voltage drop, the rate of building diminishes until at 33 amp. it has nearly stopped rising.

It would seem according to the oscillogram that the voltage wave reaches its ultimate before the field current. This is only apparent. In checking the simultaneous values of field current and voltage on the oscillogram with the load saturation curve of the exciter, it is found that the voltage is not ahead of the field current. The reason for the apparent discrepancy is that at the higher voltages, the exciter is pretty well saturated, and large changes in field current produce relatively small changes in voltage.

The advantages to be gained in using compensated exciters instead of conventional exciters are lower inductance field coils with a resulting more rapid response. For equal response for the two types, the advantage is lower field current and fewer parallel circuits in the windings. This results in less-expensive voltage regulators and greater response of exciter armature current to changes in exciter voltage due to higher ceiling voltage and lower-inductance armature circuit.

▼ ▼ ▼

Power Plant Foundations

Power plants are only as stable as the foundations they rest on. The huge sums expended in building generating stations make it imperative that this thought shall be constantly in the minds of those who design and construct them. Stressing the necessity for adequate sub-structure this article points out, broadly, the problems and their solutions.

By O. VON VOIGTLANDER
Civil Engineer, Jackson, Mich.

ANIMALS instinctively avoid dangerous footing, but man seemingly had to learn through sad experience that one square foot of Mother Earth does not have the same carrying power as another, even though covered with the same verdure.

Errors of judgment, some of them of a serious nature, were made in the construction of foundations for some of the still existing medieval churches and towers. Today, careful study and adaquate tests, interpreted in the light of the experience gained from the past, insures a maximum of safety in procedure. This is evidenced by the many existing monumental structures whose stability has been proven.

The large investment required in a power plant of modern design makes a comprehensive examination of the soil on which the project is to be erected, imperative. The construction of an adequate foundation, capable of insuring permanent stability to the entire structure, necessarily follows. Steam plants are usually located

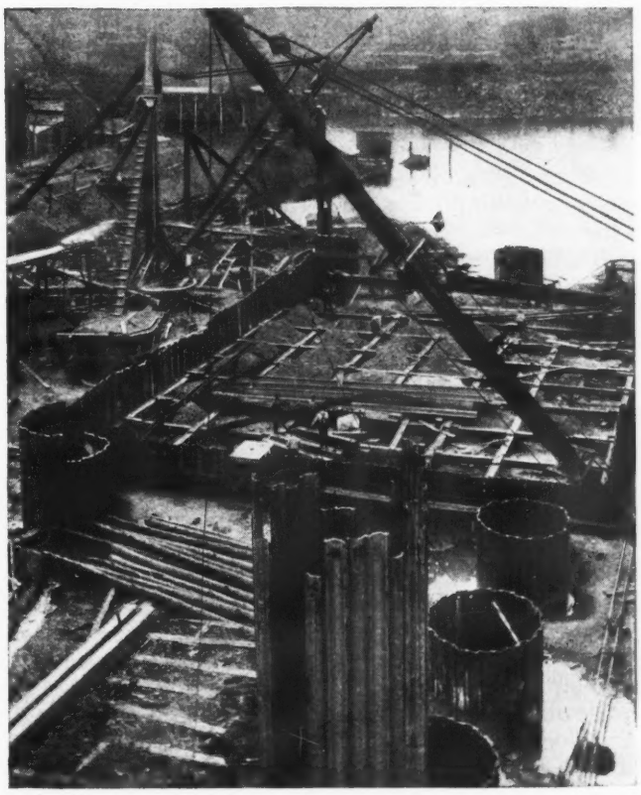

Fig. 1—Caissons to bed rock here supply support for the foundation mat

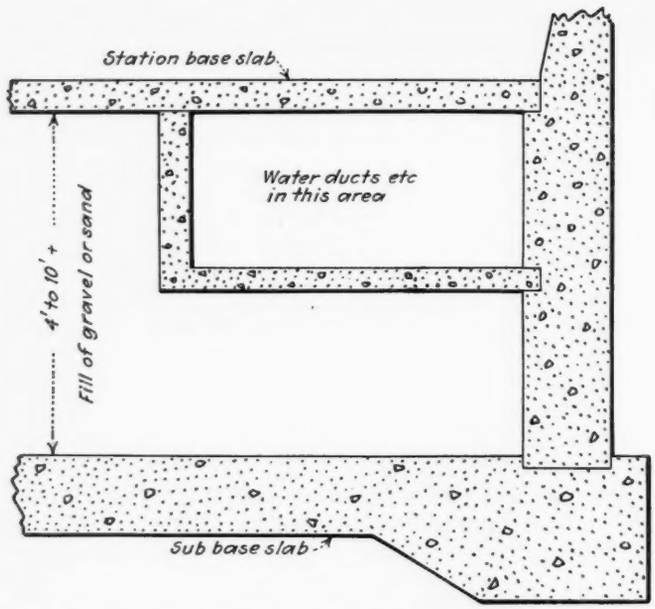

Fig. 2—Double-base slab is filled with sand or gravel for ballast

close to as large a body of running water as possible, usually on low ground, swampy soil, possibly sand, gravel, clay or a mixture of soils. Rock is met with only infrequently, and if it can be struck within a reasonable distance furnishes the ideal foundation footing.

Heavy loads placed on top of the ground will compress the soil and settle, for the material is not densely packed. If it is possible to carry the excavation down to a point where the weight of the dirt removed is about the same as the total structure load, it will be found that uniform loads placed on the substrata will not cause settlement. This depth of excavation, however, is not always possible.

Plant structures usually rest on a concrete mat or base slab which must be continuous and properly reinforced to convert the various concentrations of load arriving to it, by way of the building columns and machinery foundations, into a uniform loading at the bottom of the slab. Machine foundation slabs, even under heavy concentrations such as turbine supports, should not be separate from the main slab.

This slab in turn rests on supports which vary in nature and number according to the character of the subsoil at the site and on the existence and depth of bed rock.

First, Examine the Subsoil

Where any important structure is concerned, the first step is, therefore, a careful study of the soil formation. This soil examination comprises the digging of test pits, borings (core or wash), and actual loading of the soil at the tentative elevation of the bottom of the base slab. It is necessary to determine the location of possible underlying mine entries, submerged rivers, caverns, faults in underlying rock strata and soil stratification.

Especial attention must be given to the angle at which the various strata dip; in fact, a complete geological study should be made of the site in question. After this various information has been correctly obtained in the field and been coordinated by means of diagrams and plats, it is possible to determine the best type of subfoundation structure with a reasonable degree of assurance against disaster.

If the power house is to be located within the boundaries of a large city which has established building laws, requiring an approval of the plans by the municipal authorities, such existing ordinances may determine the type of subfoundation. The cost of this subsoil investigation work is negligible in comparison with the total cost of the project, in most cases only a fraction of one per cent.

A thick homogeneous bed of rock is the ideal subfoundation. If this exists, it is only necessary to excavate or level off the surface to a depth sufficient to allow the desired depth of basement without placing the main operating floor of the boiler and turbine room in danger of being flooded during periods of extreme high water. It is best not to place too much confidence in dykes. All main floors of the plant, outdoor substations and railroad approaches should be raised well above the highest known high water, with an additional margin of three to five feet in elevation for safety.

All rock is more or less porous at these comparatively shallow penetrations into its body. Actual tests in a number of cases have proven conclusively that it is unsafe to assume that the base slab concrete will bond into the rock and prevent hydrostatic uplift under this slab. The only safe way is to assume full hydrostatic pressure between base slab and rock surface varying closely with the variations in river elevations. It is well on important structures to make a permanent installation of test pipes penetrating the base slab into the underlying rock. These should be left properly capped off with provisions made for attaching pressure gages and, if necessary, of relieving existing excessive pressure.

Variations in river elevation may be such that to provide sufficient dead weight to resist the upward lift will require an excessive amount of concrete. In this case, fox bolts securely set into the underlying rock in sufficient number will materially save on concrete cost. In other locations it may be cheaper to use a double base slab with gravel ballast between them, as shown in Fig. 2.

If rock footings are out of the question, the resistance to loading of the underlying subsoil layers is determined. Where gravel or dense sand occurs in firmly cemented condition or in thick beds without springs or running streams, it is perfectly safe to place a mat of sufficient thickness directly on the bed. If springs exist it may be necessary to pipe these safely and permanently to a larger body of water far enough away to prevent undermining by washing. Otherwise, a sheet-pile cut-off wall or interlocking steel pile box must be installed.

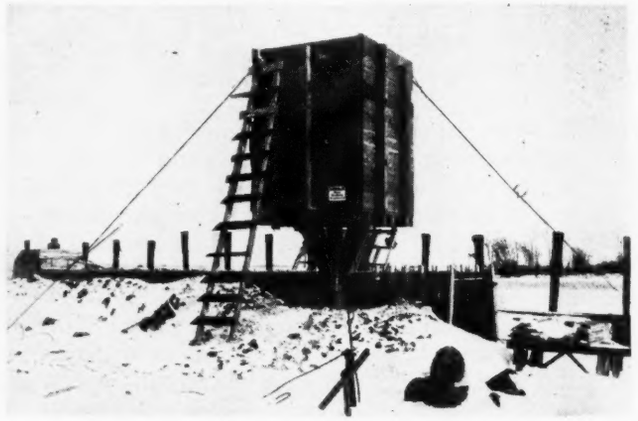

Fig. 3—Actual loading proves the carrying power of the test pile

Where softer subsoil exists, supports such as piles or caissons are used. Piling carries the superimposed loads mainly by skin friction between the pile surface and the soil material. The size and kind of timber to be used depends upon the resistance to driving and to soil condition. If the subsoil is reasonably uniform the piling may be designed by one of the skin friction formulas, but with a varying soil the safe loading of piles should be determined by formulas depending upon the penetration such as the *Engineering News* formulas for drop and steam hammers.

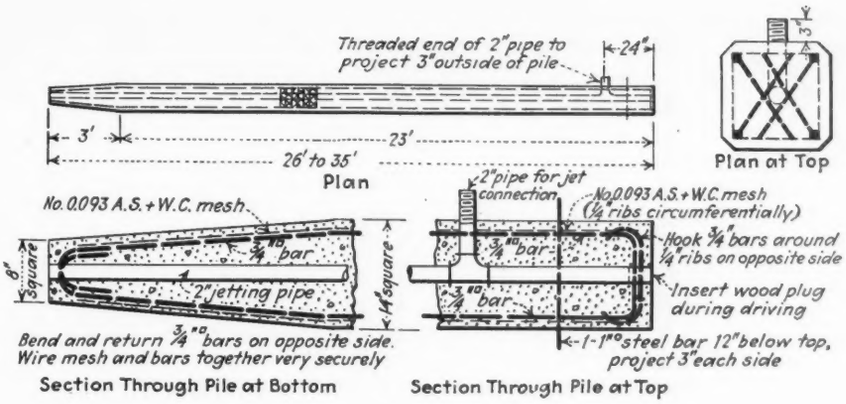
Threaded end of 2" pipe to project 3" outside of pile
24"
3'
23'
26' to 35'
Plan
Plan at Top
2" pipe for jet connection
No. 0.093 A.S. + W.C. mesh
No. 0.093 A.S. + W.C. mesh (¼ ribs circumferentially)
¾" bar
Hook ¾" bars around ¼ ribs on opposite side
Insert wood plug during driving
2" jetting pipe
¾" bar
8" Square
14" Square
Bend and return ¾" bars on opposite side. Wire mesh and bars together very securely
1-1" ø steel bar 12" below top, project 3" each side
Section Through Pile at Bottom
Section Through Pile at Top

Fig. 4—Typical design and detail of a precast concrete pile

$$L = \frac{2Wh}{S + 1.0}$$

W = Weight of hammer in pounds
h = Fall of hammer in feet
S = Average penetration per blow, (inches), of last 5 blows
L = Safe load in pounds

$$L = \frac{2Wh}{S + 0.1}$$

W = Weight of ram plus force of steam on piston (pounds)
h = Length of stroke in feet
S = Mean penetration of ten blows in inches
L = Safe load in pounds

Some adjustment in this last formula is necessary with certain types of steam hammers.

On an important structure, actual load tests must be made. After the pile has been driven to a certain loading, this load should actually be placed on the pile. Readings of settling should be taken to check against the formula, and the loading increased by 10 per cent increment up to 50 per cent over full design load. Readings are taken at each additional loading, after complete settlement, placing targets on the pile itself.

Concrete piles, either precast or cast in place, are frequently used where tops will not be submerged. Pipe columns may be used to good advantage if driven to underlying rock layers and filled with concrete. Precast concrete piles are usually made hollow to save weight. Where sluicing is possible the necessary pipes are cast into the pile. Fig. 4 shows the usual design of such piles. When such piles are driven, a wooden driving head is fitted on the concrete to avoid shatter.

Where semi-fluid muck deposited by the river is the subsoil material, caissons are the best solution, provided they rest on rock. For depths not greater than 40 ft. the open-type caisson, nothing more than a circular wall of interlocking sheet piling, is probably best. When resting on rock of sufficient bearing value, the straight shaft is filled with concrete, forming a column support. Should the sublayer lack sufficient density, increased footing is obtained by undermining the shaft bottom, increasing the footing area to any bearing value required.

Fig. 5—Conduits and drains carried in the foundation mat are shown before pouring

A caisson carries its load by end bearing only, and this must be considered in preparing the footing so that no slippage occurs. Superimposed layers of rock existing at a steep angle may necessitate penetration sufficient to avoid all danger of slippage. The air lock type of caisson is, of course, much more expensive due to the additional air lock interference. Its footage problems are the same. Tops of caissons and concrete piles are securely embedded in the walls or base slab so that the whole acts as a unit structure.

A base slab properly designed may be considered as a platform, uniformly loaded, resting on many supports, the whole being in stable equilibrium. The uniform load is the loading of the subsoil on the underside, and the supports are the building and the various equipment columns.

This slab, in addition to carrying loads safely, usually houses all necessary conduits for electric units, together with the intake and discharge ducts for circulating water. Usually all drains are also located here. Since all this must be accurately located and rigidly held during the pouring of concrete it is necessary to keep surface water out of its boundaries. Pumps placed in sumps will usually suffice. In the case of much running water, well points, of suitable size and spacing, driven around the outside of the foundation and piped to a suitable battery of pumps, will lower the surrounding watertable.

Waterproofing

It is a rather difficult job to repair a base slab if mistakes have been made. One of the most important points is to insure a water-tight structure. While a number of waterproofing compounds are on the market, it must always be remembered that the same results are being obtained by concrete mixtures using finely ground cement. A number of American manufacturers are grinding much finer cement than was customary 4 or 5 years ago.

The mixing of concrete should always be under competent engineering control. It is very important that segregation be prevented. Where concrete is spouted, a

small proportion of lime aids materially through making a smoother-running mixture. The concrete must be thoroughly spaded around reinforcing, ducts, conduits, inserts and pipes so that all voids are filled.

Even with the best of care it may be necessary to use some special treatment in order to prevent unsightly leaks in construction joints. A successful patented process consists of pouring hot asphalt into holes bored in the construction joints at suitable intervals, keeping the asphaltic matrix at as high a temperature as necessary by means of electric heating terminals previously lowered into the holes. Pressure is applied to cause this matrix to permeate all joint crevices and fissures.

It is rather difficult to construct an expansion joint so that it is water tight. In most cases practically all of the foundation in in contact with the soil, so that actual temperature changes are slight. It is possible, therefore, to design the foundation as a unit with no expansion joints and as few construction joints as possible.

On some projects the foundations up to the turbine room floor elevation were made water tight by building the foundation as boats. About four inches of concrete was placed in the excavation, after this was dry a four-ply asphalt membrane was laid (using a good grade of heavy roofing paper and hot asphalt) with the membrane projecting some five feet beyond the future base all around. The base slab and side walls were then poured and after this was dry the base slab membrane was extended up around the side walls to the top of the foundation. An additional wall four inches thick, lightly reinforced, was placed outside the membrane properly to protect it against puncturing.

The average weight of a complete steam-electric generating plant of 10,000 kw. to 35,000 kw. capacity is about 1.5 tons per sq.ft. on the total ground area covered by the plant. Except for such cases as a double-deck boiler room or turbines superimposed on several operating floors, as might occur in some extremely congested sections, the total load above the underside of the base slab will probably never exceed two tons per square foot, even for the largest units now being planned.

This loading will reach the base slab in large concentrations through the turbine, boiler and main building column supports. Closely adjacent to these large concentrations the loads coming on the base slab may be practically negligible. In fact one of the main reasons for the great thickness customary in powerhouse base slabs is the necessity to provide sufficient material to resist the punching shear of heavy concentrations and to distribute the superimposed loads over a sufficiently large area so as not to exceed the safe carrying capacity of the subsoil.

Boiler loads and most of the building column loads are quiescent. Certain other loads, however, are far from being uniform and constant. Take for example a medium-sized turbine only slightly out of balance. This unit, due to its direction of rotation and its periodicity of vibration, will give, at times, a totally different loading to its supports than that planned for a smoothly running unit. A half dozen such units, varying in their running from smooth to extremely rough will bring disastrous results unless adequate provision has been made in the design.

No cure for all ills exists, nor is it safe to say that the solution found in a certain foundation is applicable for some other project simply because general information indicates that the soil formations are identical. An actual, careful examination must be made to prove this contention before proceeding with the design and construction of the proposed structure.

Subsurface surveys sometimes show great variation of the underlying strata. It is sometimes necessary to construct costly structures covering a considerable area on several varying soils. Here the ingenuity of the designing engineer may meet some real problems. A number of structures built in recent years rest half on bed rock hundreds of feet thick, with the other half supported on foundation caissons of considerable length penetrating through to the original rock bed. When such abrupt changes are encountered, great care must be exercised to prevent uneven settling, unsightly cracks and possible complete failure of the proposed project. Carefully thought out plans conscientiously executed by the construction forces are the only safe solution.

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Providing Protection for Three-Phase Motors

By RAY BARR GLEN

FUSES are frequently used to protect three-phase induction motors against damage from single-phase current. If the motor's fuses are rated at 125 per cent or less of the current that can be continuously applied to the motor without the winding exceeding a safe temperature there is little danger of damage resulting from single-phase operation.

The safe current for a 40-deg. C. motor is 1.15 times rated full-load current. For a 55-deg. C. motor the multiplier is 1.0. Considering a 55-deg. C., 50-amp. motor the fuse rating should be $50 \times 1.25 = 62.5$, say 60 amp. Since a fuse is constructed so as to carry 110 per cent of rated current indefinitely, the motor in question can draw $60 \times 1.10 = 66$ amp. The greater the current above 66 amp. the sooner the fuse will blow.

For the same output, on single-phase, a three-phase motor will draw approximately 1.7 times normal current, or in this case $50 \times 1.7 = 85.0$ amp. This current will obviously cause a 60-amp. fuse link to melt.

Thermal cutouts are widely used to protect motors up to 20-hp. These devices open the circuit by melting a small fusible link. Larger motors are quite generally protected by overcurrent relays. These are standard relays having dashpots to provide the time delay necessary to care for momentary overloads.

Temperature relays are also in wide use for protecting large motors. These have thermal characteristics similar to those of the motors being protected; therefore safe overloads can be carried. Relays are generally connected so as to open the circuit of the motor's magnetic-switch coil or of the no-voltage-release coil.

Since a motor's copper losses vary as the square of the current, it is imperative that fuse ratings or relay settings be made as low as operating conditions will permit. Motors should be built with self-contained temperature relays, because it is the motor's temperature that is to be controlled.

Relays are available for promptly disconnecting three-phase motors in the event of single phasing, but ordinarily their installation is not justified on small motors.

An Analysis of POWER AND HEAT SUPPLY for Hotels—I

By L. E. HABBEN*

Analyzing power and heat supply for hotels, the author follows the same procedure as in his previous article, "An Analysis of Power and Heat Supply for Office Buildings," in the April 12 number of *Power*. The following are taken up in order: general considerations, electrical and steam requirements, and a typical 1,000-room hotel. First of two articles

HOTELS are classified either as residential or commercial, with of course many which cater to both types of patrons. Those investigated by the writer were high-grade commercial hotels in New York City. From the engineering point of view, the type is immaterial, except that from records it would seem that there is a falling off in the number of guests over week-ends and holidays in the commercial hotels. There also appears to be a reduction in the number of guests during the summer, especially during July and August.

When it is appreciated that a hotel may or may not have a swimming pool, house laundry, guest laundry, air conditioning, beauty shop, bake shop and a half-dozen other power and heat using elements, it is useless to employ any general formulae or curves for comparative heat-balance purposes, unless one is sure that the equipments are similar in each case; even then, the method of operation can easily produce widely differing results.

An example of the differences is shown in Fig. 1, where the electric loads for both summer and winter conditions for two New York hotels are plotted. For hotel A, hourly ampere readings were taken for one typical day each month from October to March, and those readings at the same hour were averaged and plotted to obtain the winter curve; the same procedure was employed for the months of April to September, to obtain the summer curve. For hotel B ampere readings were not available, but there was a graphic recording chart on the steam line to the engines, and the hourly steam consumptions were read off for a typical day each month and averaged as before. These four curves were then plotted to a common ordinate.

The resulting curves cannot be compared, of course, except with regard to shape, and the fact that the summer load of hotel A is higher than the winter, and for hotel B, the winter load is higher than the summer. The shape of the curves, however, is very important, and will be the subject of considerable discussion at the end of this article, when comments upon operation and the design of equipment will be made.

In the 1923 proceedings of the National District Heating Association, there appeared an interesting article dealing with the electric and steam loads found in hotels. From this the curves in Fig. 2 are reproduced. They show the percentage of the daily load occurring at each hour of the day, but no information is given regarding the equipment contained in the hotels analyzed. Instead of using these curves, therefore, it will be more instructive to build up curves for a typical hotel and compare them with Fig. 2 afterwards.

Electrical Requirements

In the previous article on office buildings, the writer suggested a value of 1.5 watts per square foot of floor area as being a fair value for the electrical load, but no such general figure can be used for hotels. Although the following may seem at first to be a series of assumptions, it will be interesting to follow them through, afterwards consider the probable error in so doing, then observe how they check with actual practice.

The electrical power requirements can be estimated fairly accurately when the connected motor load is known, except perhaps for elevators and laundry motors. A 10-hp. motor will mean 7.5 kw. back on the source of supply. The calculations are based upon 80 per cent full load, 86 per cent efficiency and a loss of 7 per cent between the motor and the source of supply. Most of the larger motors for ventilating, refrigeration, vacuum system, etc., will run for fairly long periods and should approach this very closely. The service lighting will be in the order of 0.3 watts per square foot of floor area. Guest room lighting will be in the order of 0.5 watts per square foot of floor area, although this is open to wide fluctuation due to the differences in types of lighting fixtures and the interior color schemes. This is the most uncertain factor of all.

The following were taken from an actual hotel which kept the most complete records of any visited, and lists the equipment, horsepower, kilowatts on the electric supply

*Formerly with Lockwood Greene & Company, McClellan & Junkersfield, Ford Bacon & Davis, and the Electric Bond & Share Company.

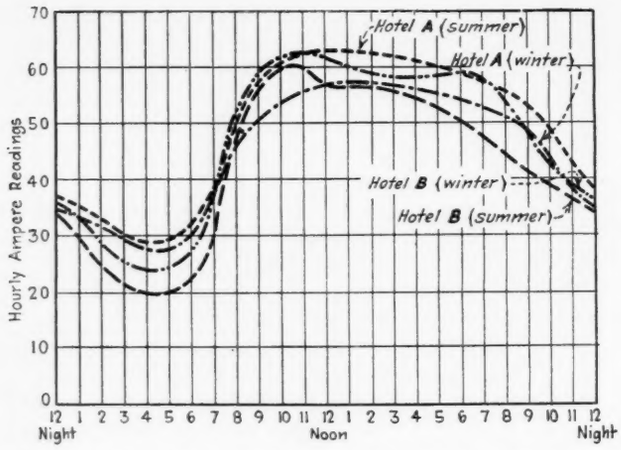

Fig. 1—Form of hotel electrical loads, summer and winter

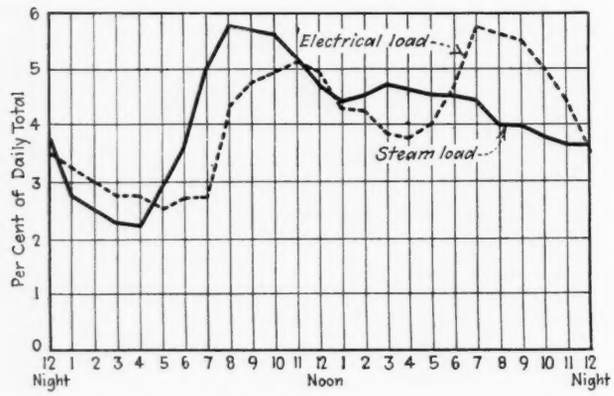

Fig. 2—Variation of steam and electrical demands, from N.D.H.A. proceedings

system, also the approximate hours of operation. These are plotted in Fig. 3.

	Hp.	Kw.	Operation
Engine room and toilet vents	70	52½	12 a.m. to 12 a.m.
Hot water pump	10	7½	12 a.m. to 12 a.m.
Service lights	..	45	12 a.m. to 12 a.m.
Refrigeration	53	40	12 a.m. to 12 a.m.
Grill room	..	10	6 a.m. to 1 a.m.
Kitchen	..	25	6 a.m. to 1 a.m.
Vacuum cleaner	25	19	1 a.m. to 5.30 p.m.
Dining room	..	10	7 a.m. to 12 a.m.
Lobby	..	20	7 a.m. to 12 a.m.
Basement	20	15	8 a.m. to 5 a.m.
Laundry	..	65	8 a.m. to 5 a.m.

The laundry curve is cut back because washing would begin before finishing, and finishing would continue after all the washing was complete. So far the only thing open to question is service lights, which comprise halls, corridors, stair-wells, etc.

The hotel was twenty stories, 8,300,000 cu.ft. total or approximately 415,000 cu.ft. per floor. An average floor height of 10 ft. gives 41,500 sq.ft. per floor. Service space will be approximately 20 per cent of this, or 8,300 sq.ft. per floor, but the ground floor and two basements were occupied by the lobby, dining room, ball room, grill, laundry, kitchen, engine room, etc.; so that only seventeen floors are figured for service lighting, which works out as follows:

$8{,}300 \times 17 \times 0.3 = 42$ kw., say about 45 kw., because the lobby will be lighted 24 hr. a day (partially lighted during the early morning).

The first curve above these plotted values on Fig. 3, is the guest lighting, arrived at as follows:

With a total area per floor of 41,500 sq.ft., of which 80 per cent is guest floor area, the total guest lighting load will be $33{,}000 \times 17 \times 0.5 = 280$ kw., if the rooms were 100 per cent rented and all the lights burning. Assuming the rooms 70 per cent rented, and that only one-half of the lights will be in use at any one time because of the diversity in time of rising in the morning, this lighting load becomes 95 kw., and the same value has been used for the evening. During the day some rooms will be occupied, also the cleaning of guest rooms will be undertaken, so a figure of 65 kw. has been used for the midday period. It will be noted that the effect of daylight has been neglected.

The curve at the top of Fig. 3 gives the average elevator load, taken from tests upon a bank of six gearless, voltage-control, passenger elevators, where the maximum five-minute period averaged 31 kw. Inasmuch as there were eleven elevators in the hotel, this average load has been taken to be 60 kw.—the peak periods have been

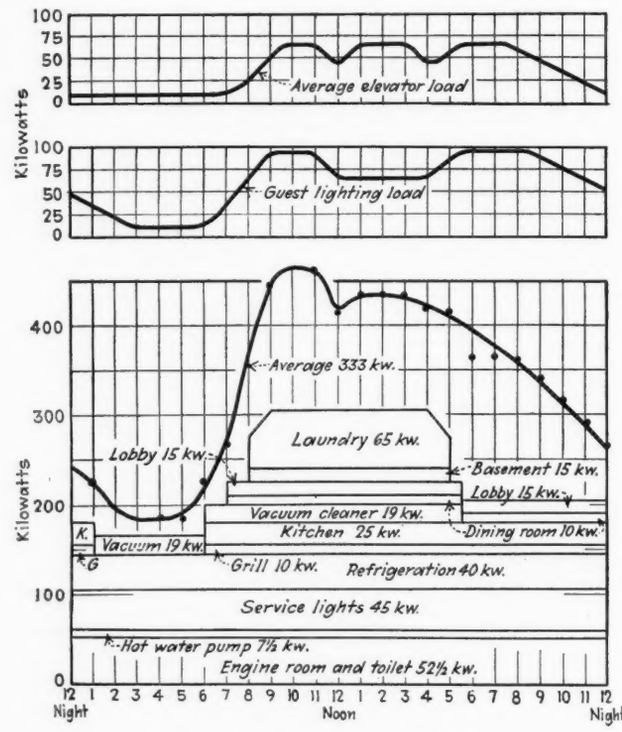

Fig. 3—Electrical loads, hotel B

assumed. By plotting the guest lighting and the elevator loads upon the bottom figure, a series of points are obtained through which a curve has been drawn. It gives the average electrical load.

Reviewing the foregoing data, the only thing that is open to serious question is the guest lighting. The service lighting may vary in amount by a kw. or so, but it will not alter the shape of the curve. The elevator load has been taken as stated, from actual tests with a graphic chart upon a bank of six machines. The rest of the curve cannot alter very much except as a result of changes in operation.

As this synthetic graph has been built up from data on hotel B whose load curves, taken from the steam chart, are plotted on Fig. 1, the shape of these curves may be compared. They are plotted together to a common ordinate on Fig. 4 and show a remarkable similarity. Not only does the shape of the built-up graph correspond

closely with the curves obtained from the steam chart readings, but also if the kilowatt hours are calculated from Fig. 3, they are found to be 8,000 kw.-hr. per day, or 240,000 kw.-hr. per month. The actual metered consumption for 1928 was a maximum of 263,107 kw.hr. for January and a minimum of 213,470 kw.-hr. for July.

It will be noted that a few intermittent motors, such as the cold water house pump, air compressor and sump pump, have been omitted. These could be included in the built-up graph by adding the average kw. per hour during the day. They would probably consist of a 50-hp. cold water pump with four starts per hour, a 10-hp. sump pump motor and a 15-hp. air-compressor motor. Naturally, special functions in the private dining room or ball room will add to the load requirements as estimated and alter the shape of the curve. This probably accounts for the peak in the winter curve of hotel B around 6 o'clock on Figs. 1 and 4.

Steam Requirements

The heating requirements for a hotel are more exacting than those for an office building, because guests demand heat at all times of the day and night. Therefore, the factor of $5\frac{1}{2}$ lb. of steam per cu.ft. for the latter must be increased to $8\frac{1}{2}$. The total heating steam for one season will be calculated by multiplying the cubic contents of the building by $8\frac{1}{2}$. This total will be distributed as follows, in per cent; October 6, November 8, December 16, January 20, February 20, March 15, April 10, and May 5.

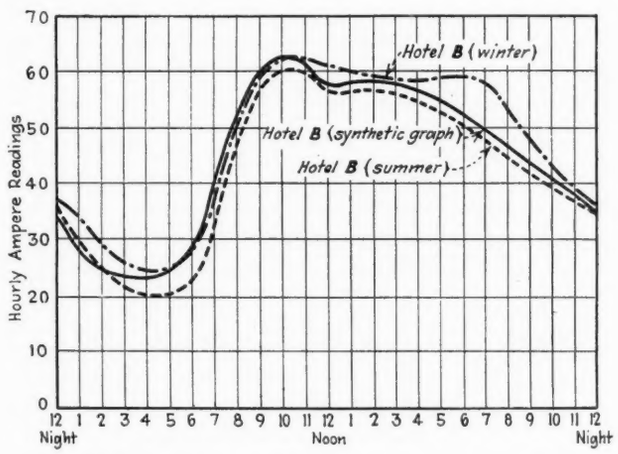

Fig. 4—Showing similarity between synthetic graph and curves for Hotel B

The hot water requirements for the house, taken from a hotel where they were metered, varied between 306 and 560 lb. per person per day. For the typical hotel, 500 lb. per person per day will be taken.

Accurate data concerning the hot water and steam required for the kitchen were most difficult to obtain because of the unlimited variety of equipment possible, so the figures given are based upon a similar hotel where these quantities were actually measured by meter. The steam and hot water for the typical laundry were kindly supplied by one of the largest manufacturers of laundry equipment.

The refrigerating load for ice water, kitchen and ice-making will be considered a steady one for 24 hr. per day, because of the necessity of making clear ice, which takes from 24 to 30 hr. As mentioned previously, there is a falling off in the number of patrons during the summer months, but the quantity of ice and hot water seems to be fairly constant.

Typical 1,000-Room Hotel

The typical hotel will be assumed to have a volume of 6,000,000 cu.ft. and to be eighteen stories high, with a total floor area of 600,000 sq.ft., or 33,000 sq.ft. per floor. The average number of guests is assumed as 800 for the eight winter and fall months and 600 for the four summer months. Before starting to develop curves, the following equipment will be assumed:

Refrigerating machinery—one 30-ton machine for air-conditioning of dining rooms and grill; one 50-ton machine for ice making, kitchen, ice water, etc.

Laundry. For house and guests.

Kitchen. For grill, dining and private dining rooms.

Cleaning. Vacuum system for corridors, guest rooms and public places.

Ventilation. Laundry, basement, dining rooms, grill, lobby, ball room, private dining rooms, kitchen, toilets, engine room, etc.

Air conditioning. Grill and dining rooms.

Electrical Loads

Three floors will be devoted to lobby, dining rooms, ball room, grill, laundry, engine room, etc., so the service lights will amount to $0.2 \times 33{,}300 \times 15 \times 0.3 = 30$ kw. Guest lights will be $0.8 \times 33{,}300 \times 15 \times 0.5 = 200$ kw.

If the hotel is 70 per cent occupied and one-half of the lights are in use at one time, the load on the electric supply will be 70 kw.

The connected loads (horsepower and kilowatts at the source of supply) and the hours of operation are listed in the table at the top of the following page.

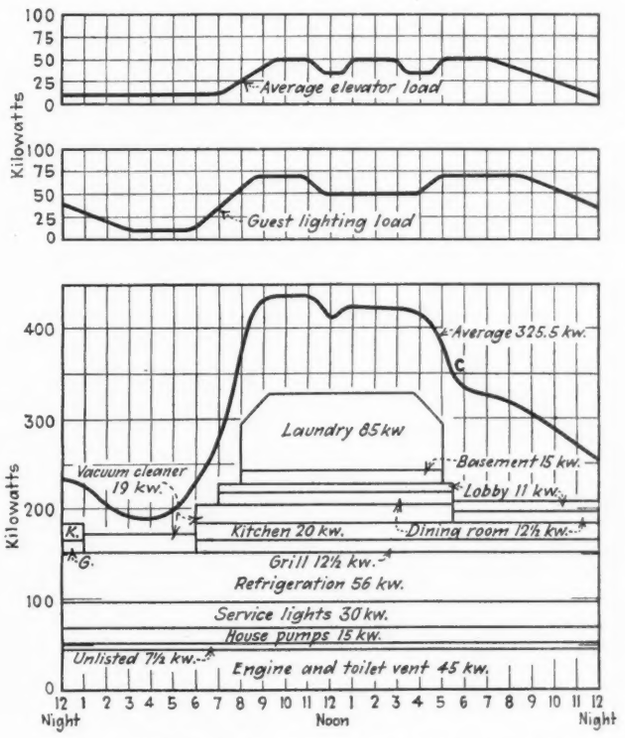

Fig. 5—Electrical loads for typical 1000-room hotel (October to May inclusive)

	Hp.	Kw.	Operation
Engine room and toilet vents	60	45	12 a.m. to 12 a.m.
Unlisted (intermittent)	10	7½	12 a.m. to 12 a.m.
House pumps (intermittent)	20	15	12 a.m. to 12 a.m.
Service lights	..	30	12 a.m. to 12 a.m.
Refrigeration (kitchen, etc.)	75	56	12 a.m. to 12 a.m.
Refrigeration (air conditioning)	60	45	7 a.m. to 1 a.m.
Grill	..	12½	6 a.m. to 1 a.m.
Kitchen	..	20	6 a.m. to 1 a.m.
Vacuum cleaner	25	19	1 a.m. to 5.30 p.m.
Dining room	..	12½	7 a.m. to 12 a.m.
Lobby	..	11	7 a.m. to 12 a.m.
Basement	20	15	8 a.m. to 5 p.m.
Laundry	145	85	8 a.m. to 5 p.m.
Guest lighting (max. load)	..	70	.....
Elevators (average max.)	..	50	.....

These values are plotted as curve C, Fig. 5, which indicates the expected load for 245 days per year, that is, from October to May. This load is found to be 7,800 kw.-hr. per day or 1,910,000 kw.-hr. for 245 days.

For the summer months, June to September, the load will increase by the amount of refrigeration used for air conditioning, but the laundry, elevator and guest lighting loads will decrease in proportion to the number of guests. These loads are plotted as curve D, Fig. 6, and are found to be 8,140 kw.-hr. per day or 976,800 kw.-hr. for 120 days.

If this yearly consumption of 2,886,800 kw.-hr. were purchased according to a schedule similar to the New York Edison Company's service classification No. 5, the annual electric power bill would amount to $57,168.*

Steam Requirements

The steam requirements will be 8½ times 6,000,000 = 51,000,000 lb. per year. The four winter months will average 18 per cent of this, and the four fall and spring months will average 7 per cent, or:

51,000,000 times 0.18 times 4 = 36,700,000 lb. for December to March.

51,000,000 times 0.007 times 4 = 14,300,000 lb. (approximately) October, November, April and May.

The hourly distribution of the heating steam for December to March is plotted as curve J, Fig. 8. The hourly distribution of the heating steam for October, November, April and May, is plotted as curve L, Fig. 9. Figs. 8 and 9 will appear in the second section of this article, to be published next week.

Low-Pressure Steam

Hot water for the house will be 500 × 800 = 400,000 lb. per day of 24 hr., which will require approximately 50,000 lb. of steam per day. The hourly distribution of this steam is shown as curve E, Fig. 7; the shape of the curve is according to a similar hotel which had a graphic chart on the steam for the heating of hot water.

The hot water for the kitchen will be 360,000 lb. per day of 20 hr., which will require approximately 45,000 lb. of steam per day. The hourly distribution of this steam is shown as curve F, Fig. 7. It has been assumed that there is a peak after each meal.

The hot water for the laundry will amount to 190,000 lb. per day of 10 hr., which will require approximately 24,000 lb. of steam per day. The hourly distribution of this steam is indicated on curve G, Fig. 7. It has been assumed to have a rapidly rising front, due to the starting of washing apparatus, a low point when the help is at lunch and a gradual falling off as the washing is finished ahead of the drying and ironing. The sum of curves E, F and G gives curve H, which is the low-pressure steam demand for hot-water heating during the months October to May.

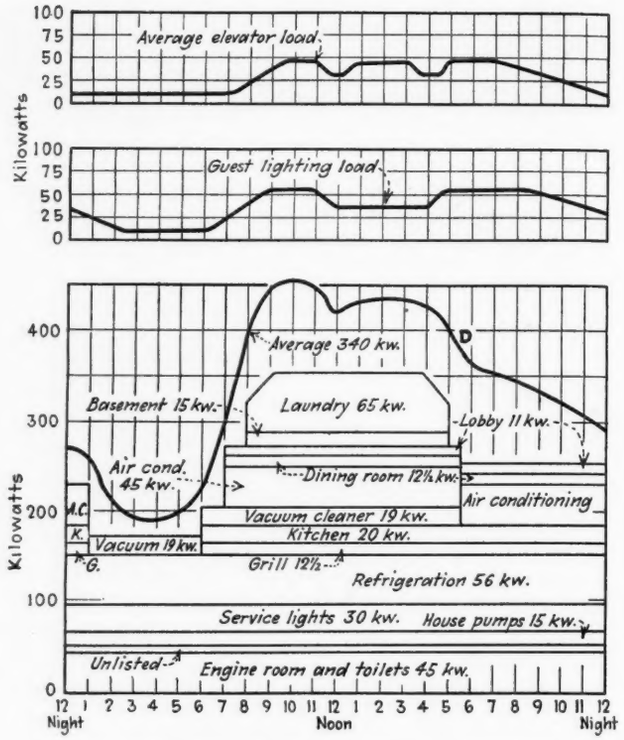

Fig. 6—Electrical loads for typical 1000-room hotel (June to September inclusive)

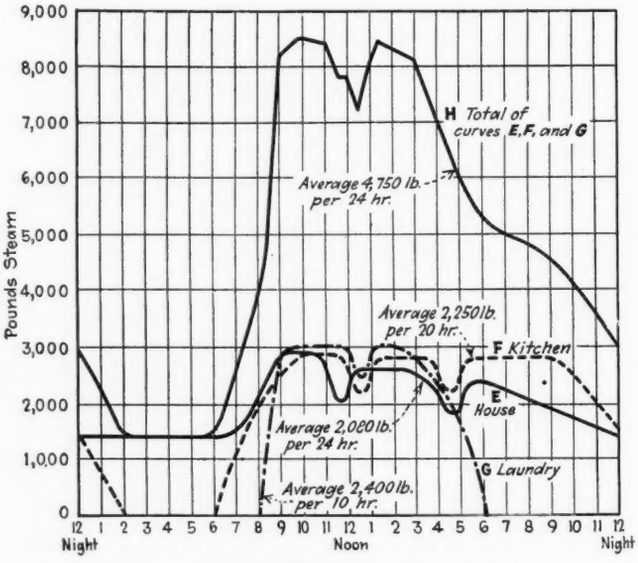

Fig. 7—Steam for hot-water heating (October to May inclusive)

* The New York Edison Co's Wholesale Classification No. 5 is as follows:

a. Demand Charge,
For the first 300 kw. of maximum demand, $2.50 per kw. per month
For any excess over 300 kw. of maximum demand, $1.50 per kw. per month.

b. Energy Charge,

	Kw.-Hr. per Month	Cents per Kw.-Hr.
For the first	12,500	2½
For the next	50,000	2
For the next	50,000	1½
For the next	50,000	1
For the next	500,000	.9
For the next	500,000	.8

Minimum guarantee shall not be less than 800 kw. or $1,500 per month, exclusive of energy and coal charges.

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High-pressure steam requirements, comparative purchased steam and boiler plant set-ups, and a general discussion of operation and design are included in the concluding installment, to be presented next week

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Crank Pin Pressures In Gas Engines

By G. GROW

Contrary to a prevalent belief among engineers operating gas or oil engines, the crankpin bearing seldom, if ever, pounds. Before reducing bearing clearance the engineer should be certain that this adjustment is necessary. This article shows that crankpin bearings do not pound and that pounding, if it exists, is at the piston pin

SOME time ago the author had occasion to inspect a certain plant containing six vertical gas engines having 12x18-in. cylinders and operating at 200 r.p.m. The engineer was busy removing shims from the crankpin-end bearing of one of the connecting rods. To the inquiry of why the adjustment was being made, the engineer replied that "he just couldn't keep the engines from pounding at the crankpin."

Upon suggestion that a crankpin bearing seldom if ever pounded, he strode to an order blackboard and drew four sketches similar to those in Fig. 1. He indicated the points on the bearings where the pins made contact during the four strokes, and declared that it was evident that at the end of the exhaust stroke the pins moved to the opposite sides of the bearings, causing pounds, and that the same thing occurred at the end of the suction and the start of the compression strokes. Too large clearances would cause pounding, so he was removing shims.

This engineer's belief is shared by so many operators that an outline of what does occur at the crankpin and wristpin of a four-cycle engine justifies itself. The following is the gist of my discourse to the engineer.

While it is not quite safe to say that a crankpin bearing never pounds, nevertheless crankpin pounding seldom occurs. The effect of the inertia of the reciprocating parts upon the bearing pressures of an oil or gas engine is not always grasped by the operating engineer. As a consequence he may reduce the clearance of a crankpin bearing to eliminate a wrongly interpreted pound or he may attempt to locate a pound by checking the piston that is starting its suction or compression stroke, and will arrive at the wrong conclusion.

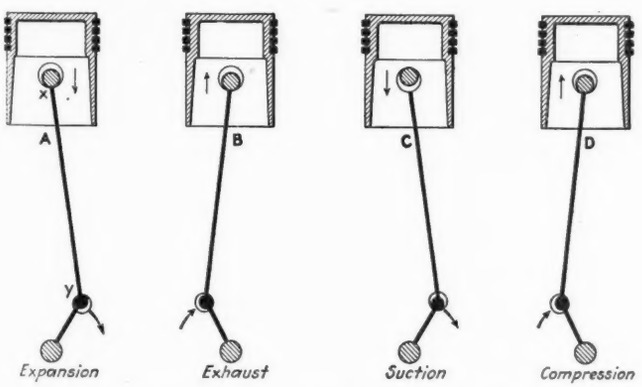


Fig. 1—Analysis of pin pressures, ignoring inertia forces

The effect of the inertia, or the resistance to a change in velocity, of the reciprocating parts actually shifts the crank position where the bearing pressures are reversed, if not eliminating the shift altogether.

In Fig. 1 is an outline of the points when the piston pin and crankpin would contact with the bearings if only the gas pressure within the working cylinder acted upon the piston. It will be observed that from the beginning to the end of the expansion stroke, Fig. 1 *A*, the two pins are in contact with the inside halves of the bearings at *X* and *Y*. When the power stroke is ended and the exhaust stroke begun, Fig. 1 *B*, the crankpin forces the connecting rod and piston upward, thereby maintaining contact between the pins and the inner halves of the bearing. But as the piston reverses and starts upon the suction stroke, the crank must pull the rod and piston downward, and the two pins at once shift the contact to the other side of the bearings, as shown. On the compression stroke, the crank must force the rod and piston toward the cylinder head, consequently the pin pressure shifts from the outer to the inner halves of the bearings

Inertia of the reciprocating parts, however, upsets these assumed conditions.

Force is necessary to increase or decrease the velocity of the reciprocating parts. With the piston at one end of its stroke, force must be exerted on these parts to bring them from a state of rest to the maximum velocity, and likewise a force is needed to slow the parts as the piston approaches the opposite dead center. To start the reciprocating parts, some of the pressure exerted by the cylinder gases is used, and in slowing down the parts exerts a force on the piston pin and through the connecting rod upon the crank.

The quantity of this force depends upon the weight of the reciprocating parts, upon the crank velocity and upon the ratio of the connecting rod to the piston stroke. It varies at each crank position. Expressed in a formula, the inertia force,

$$F = \frac{WV^2}{gr}\left[\cos a + \frac{\cos 2a}{n}\right]$$

When a = crank angle;
r = crank throw in feet;
n = rod to crank ratio;
V = velocity of crankpin;
W = weight of reciprocating parts;
g = acceleration due to gravity, 32 ft. per second per second

If the value of this inertia force be plotted for various crank angles, with the weight W of the reciprocating parts known, the inertia-force curve FG, Fig. 2, is obtained for the expansion stroke; the inertia curves for the suction stroke is identical, while those for the ex-

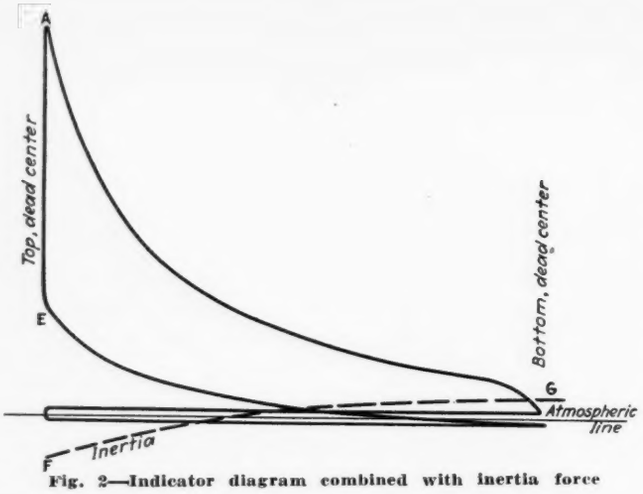


Fig. 2—Indicator diagram combined with inertia force

haust and compression strokes are identical but reversed.

When the curve is below the zero pressure line, the force is negative, that is, force must be applied to accelerate the parts, and when above the zero line the slowing up, or negative acceleration, gives up a force which augments that exerted by the cylinder gas pressure upon the piston.

The cylinder indicator diagram and the inertia force diagram may be combined to give the diagram of the net pressure exerted upon the piston during the four strokes. This has been done, using the indicator diagram and inertia force diagrams of Fig. 2, to form the net pressure diagram of Fig. 3. In this figure the four strokes have been placed end to end. Positive pressures, that is, pressures exerted on the crank, are above the zero line, and negative pressures, or the forces exerted by the crank upon the rod, are below. The points of contact of the piston pin with its connecting rod bearing are shown for the four strokes, with the clearance greatly exaggerated.

It will be seen that at the position in the piston stroke where the net pressure line crosses the zero line, the point of bearing contact shifts from inner to outer, or from outer to inner side of the piston pin bearing. This shift does not occur at the end of the strokes, as is commonly assumed by operators, but practically in the middle of the suction and exhaust strokes. It is at these points that pounding at the piston pin will occur if too much clearance exists. The pressure reversal occurs at these points in any event.

The amount of the pound depends upon the bearing clearance, the piston speed and the amount of pressure change per unit time. From the instant the pin leaves contact with one side and makes contact with the other side of the bearing, the reciprocating parts are free bodies moving at a definite velocity. If the rate of pressure change is high, the pin delivers a heavy hammer blow to the bearing. A high rate of pressure change may be due to a high inertia force, but either heavy parts or a high piston speed, which is the reason why high-speed engines tend to pound more than do slow-speed units. Or excessive clearance may permit a considerable change in pressure to occur between the times the bearing contact is broken and made again on the opposite side.

It is possible, however, for the compression line to be of such shape that when the inertia force curve is combined with it, the net pressure curve cuts the zero line at two points in the piston stroke. In this event a pound may occur at these points of pressure reversal. Likewise if a gas engine is operating at low load on a weak mixture, pounding may occur at the start of the power stroke.

Furthermore if the engine is lightly loaded and the cooling water is allowed to run freely, the cooling of the gas-air mixture may cause the compression line to be so low that the inertia force is greater than the negative compression force at the end of the stroke. In this event at the start of the power stroke a pressure reversal occurs at the piston pin, producing a pound.

It is to eliminate this pound at the end of the compression stroke that designers go to high compressions on high-speed engines, for the high speed increases the inertia force which must be offset by a negative compression force.

With two-cycle engines there is no reversal of the bearing pressure, so no piston pin pound should occur. It is possible, however, for pounding to occur if the compression line is too low, assuming either heavy reciprocating parts or a high piston speed. For under these conditions the piston may be pulling the rod at the end of the compression stroke, with the consequence that a reversal of pressure occurs at the beginning of the power stroke. Such pounding can be eliminated or reduced by delaying the ignition, so that the pressure reversal occurs when the piston has proceeded on its stroke.

In this discussion the gravity effect of the reciprocating parts have been ignored for the resulting acceleration force due to gravity is so small compared to the inertia and gas pressure that its influence may be ignored.

These remarks all pertain to the piston pin. The conditions at the crankpin bearing are entirely different.

Before examining the pressure events at the crankpin it is advisable to study the origin of the force that is exerted along the connecting rod. This force is made up of two components, the net force exerted on the piston, which has already been discussed, and the reaction perpendicular to the cylinder axis. In a cross-

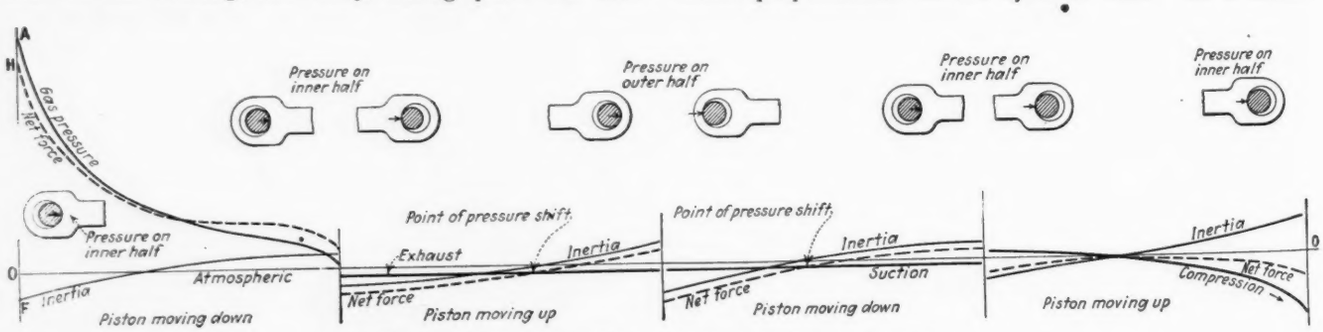


Fig. 3—Diagrams showing net pressure acting on piston pin

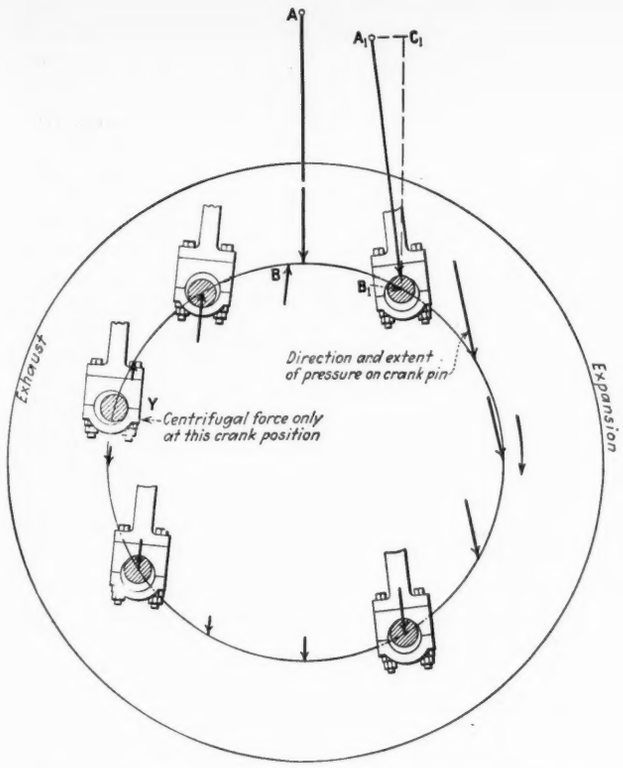


Fig. 4—Diagram showing direction and magnitude of pressure on crankpin during expansion and exhaust

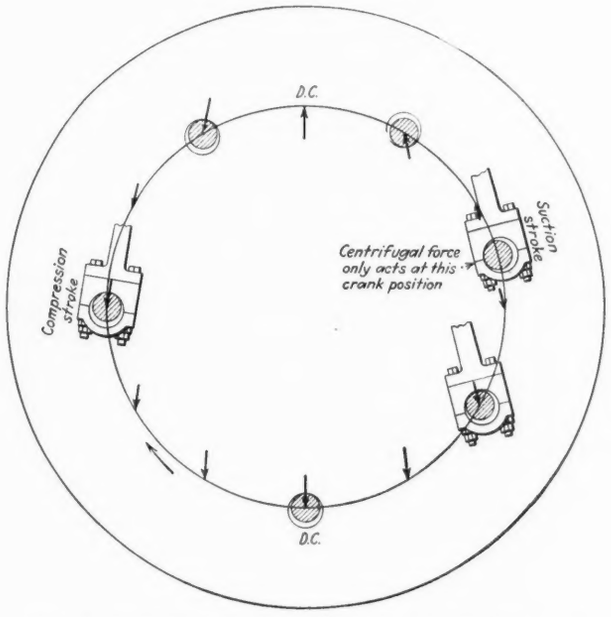


Fig. 5—Diagram showing direction and magnitude of pressure on crank during suction and compression

head type engine this is usually termed the "guide reaction." The force acting along the rod at any point in the stroke may be obtained either mathematically or by laying off the value of the net piston pressure and constructing a force parallelogram, as in Fig. 4, where B_1C_1 is the net piston pressure, C_1A_1 the side reaction and A_1B_1 the force along the connecting rod at the 30-deg. crank position.

In obtaining the net piston pressure, one must take into consideration the reciprocating motion of the rod, and construct the inertia force curve to be applied in finding the force acting along the rod. The connecting rod has two motions; one a reciprocating motion in the direction of the cylinder axis and a curvilinear motion with the crankpin as the center. Practically the same net effect is obtained if it be assumed that one-third the rod weight be centered at the piston pin and two-thirds be placed around the crankpin. It is necessary to add this one-third rod weight to the piston and piston pin weights to establish the inertia force due to the reciprocating parts that acts along the rod. By combining this new inertia force curve with the indicator diagram, the net force acting upon the rod in a direction parallel to the cylinder axis may be found.

It is no great matter to establish this net pressure for various crank positions. But this force along the rod is not the only force acting between the crankpin and its bearing.

The two-thirds of the rod weight assumed to be centered about the pin exerts a centrifugal force which tends to cause the crankpin and rod bearing to contact on a radial line. The value of this centrifugal force is obtainable from the Equation $F = \frac{WV^2}{2\ gr}$ when W is the weight, V the crankpin velocity in feet per second, g the acceleration due to gravity of 32.2 ft. per second per second and r the crank radius in feet. Compared to the reciprocation inertia force this is small and is of moment only when the crankpin pressure shifts from one side of the bearing to the other.

At any crank position, then, the actual force exerted between the pin and its bearing, and the direction of the force, is the algebraic sum of the force along the rod and the centrifugal force. Obviously at 30 deg. on the expansion stroke the contact between the pin and bearing is at B_1 as shown in Fig. 4.

Reversal of Pressure Occurs Midway of Exhaust and Suction Strokes

Bearing in mind that it has already been shown that there is a reversal of piston pressure at about midway the exhaust and suction strokes, obviously it is at these points in the crank travel that a reversal of the crank pin pressures is possible, and it will be shown that even at these points no sudden crankpin pressure reversal occurs.

In Figs. 4 and 5, several crankpin positions are shown for the four strokes for which the direction and value of the crankpin bearing pressures have been found as discussed. It will be seen that the point of contact between the crankpin and its bearing is on one side of the bearing until the point Y in the exhaust stroke, Fig. 4, is reached. At this point the force along the connecting rod becomes zero, but due to the centrifugal force exerted on the revolving end of the connecting rod the pin contacts with the bearing as shown at Y in Fig. 4. Consequently the pin does not move instantaneously from one half of the bearing to the other but slides along the babbitt. The same shifting occurs on the suction stroke where the only force acting is centrifugal.

It will be seen that at no point does a complete reversal occur. Consequently there can be no pound at the crankpin bearing, although this statement may not apply in event of abnormal conditions.

In conclusion it can be stated with safety that bearing pounding in a gas or oil engine only occurs at the piston pin.

Routine Inspection Schedule for Electric Motors and Control

By MARIN PHILLIPS

MANY schedules for inspecting electrical equipment have been devised and applied with varying degrees of success. The one suggested in this article has proven highly satisfactory in a large plant, and its use has resulted in a very large reduction in maintenance costs and outage time of equipment. A section of the motor and control inspection chart is shown, and from this a complete chart may be worked out to cover a year's records of inspections for any group of motors and their controllers. The procedure of making the inspections is given in the following:

INSULATION RESISTANCE: The resistance of the circuit to ground includes the control equipment end of the line switch, wiring, control equipment, wiring to motor and the motor winding. The resistance is measured with a megohm meter by connecting one test lead to ground and the other to the dead end of the line switch. In the case of a starting compensator it is placed in the starting position and a reading taken, and then thrown over to the running position and another reading taken. This test is made with the oil pan in place. Any part of the circuit found with a resistance of one megohm or less is given immediate attention to rectify the trouble.

CONTACTS: All burned out contacts are changed or filed to remove burrs and to arrest further burning and poor contact. Contacts that have been filed considerably are used on the starting side only. When the contacts are changed that are not too badly burned, they are filed and dressed for future service. Contact fingers with weak or broken springs are also changed. The condition of the undervoltage release coil is noted at this step of the inspection.

RELAYS: Current- and time-limit settings are checked. Relays are not set for more than a 25 per cent overload, except in an emergency and special cases. Oil dashpots are inspected for free operation, and the relay contacts must reset when raised. A close watch is kept for charred insulation on coils and connections denoting overheating, and the tripping circuit is inspected for loose or poor contacts.

LINE SWITCHES: Inspection is made for worn parts, hot and loose connections. Conditions of the cross-bar insulating tubes are noted, and if they show signs of cutting or deterioration, or look unsafe, they are replaced with new ones. Switches must be working freely, so that they will open and close easily without forcing their handles. If switch contacts do not open with a quick break, or close completely by the action of the actuating spring, the contacts are given a thin coat of vaseline and the working parts oiled. At this inspection, the condition of the switches is noted as to repairs, safety and painting. Care is taken to have the switch cabinet doors closed, if the switch is alive when it is opened or closed. After a switch is closed the door is then opened to see if the contacts are fully closed.

CLEANLINESS: The switch and control equipment are cleaned inside and out. All dirt, carbon and copper dust are wiped off the finger and contact boards in the starter. All sediment in the bottom of oil pan is cleaned out and the condition of the oil is noted for water.

SAFETY: All ground wires are inspected to see if any are disconnected or broken. All wiring is inspected for bare wires, poorly taped connections and insulation abrasions. Where open wiring enters or leaves the metal cabinet, switch boxes, etc., it is inspected to see if it is protected by suitable insulation bushings. The insulation on all contact support bars in the compensators are watched for signs of cutting and deterioration that would result in a ground or short circuit.

All wiring in starters and control equipment is inspected to see that none bears too heavily on conduit bushings, sharp metal corners and the like, that would cut through the insulation, also, that no wiring rests on any moving parts to wear the insulation. Care is taken to see that switch doors and starter covers are in place when starting motors. The oil level in the oil pan is noted to see that it covers the contacts. Care is taken to see that line switches are open before starting any inspection or repair work and that danger signs are used.

	January												February											
Motors	Insulation and Resistance	Contacts	Relay	Switch	Cleanliness	Safety	Fuses	Brushes	Belt and Pulleys	Bearings	Air Gap	General Condition	Insulation and Resistance	Contacts	Relay	Switch	Cleanliness	Safety	Fuses	Brushes	Belt and Pulleys	Bearings	Air Gap	General Condition
Power House																								
Boiler House																								
Coal Handling																								
Refrigeration Plant																								
Air Compressors																								
Filter Plant																								
Pumps																								
Elevators																								
Ventilation																								

Section of a motor and control inspection schedule chart

FUSES: Fuses are inspected for proper size, contact heating and charred cases. Fuse clips and fuse contact blades and ferrules are cleaned when found dirty or corroded.

MOTOR INSPECTION: Condition of the brushes is noted, they are changed when necessary and the spring pressure is checked. Brushes are raised and lowered in the holders a few times to see that they are working freely. This also gives a check on the shunts, which will reveal if they are loose in the brushes or terminals. Notice is to be taken to see if the brushes line up properly on the commutator or slip rings.

BELTS: The general condition of the belts is noted, such as wear, coming apart, hooks parting, too much or not enough belt dressing, or belt indicating slipping. Condition of the pulley is noted whether loose or badly worn, or broken if paper.

BEARINGS: Bearings are gaged for wear and inspected for oil leaks around the oil gage, drain plug and shaft. The condition of the felt washers on the inside of the bearings is noted and oil gages are cleaned when dirty.

GENEAL CONDITION: Ground wires are inspected and the condition of the other wiring noted. Inspection is made to determine if water is dripping on the motor. The condition of the windings is examined to note any effect of dirt or oil, and if they need painting, also to see if any other repairs are needed.

PRACTICAL AIDS TO OPERATION

Construction of the Flue Gas Analysis Chart

THE flue gas analysis chart illustrated in Fig. 1 appeared in connection with an article in the May 3 issue of *Power* entitled "Regulating Natural Gas Fires for Best Efficiency" written by the author. This chart created interest among readers of *Power*, some of whom wished to construct a similar chart for a gas radically different from the north Louisiana gas used. The author therefore presents the method of construction.

THE PERFECT COMBUSTION LINE

The perfect combustion line is established by first determining the maximum possible CO_2 content of the flue gas. This condition is attained with zero excess air and is determined in the following manner:

Fuel constituents— mols / Products of combustion— mols

$$0.965C\,H_4+0.965\,(2)\,O_2+0.965\,(2)\,(3.78)\,N_2 = 0.965\,CO_2+2(.965)\,H_2O+7.3\,N_2$$
$$0.021\,N_2 = 0.021\,N_2$$
$$0.014CO_2 = 0.014CO_2$$

$$1.0\ \text{gas}+(1.93O_2+7.3\,N_2) = 0.979CO_2+1.93\,H_2O+7.321\,N_2$$

Fuel + (Air) = Products of combustion

The Orsat analysis gives the analysis of the dry products of combustion, therefore the maximum percentage of CO_2 will be

$$\frac{0.979CO_2\times 100}{0.979CO_2+7.321\,N_2}=11.8\ \text{per cent}$$

This value of CO_2 establishes the upper corner of the chart and also is one end of the zero excess air line. The lower right-hand corner of the chart gives the analysis of dry air, that is no CO_2 and 20.9 per cent O_2. The perfect combustion line is drawn between these two points.

SLOPE OF THE EXCESS AIR LINES

The other end of the zero excess air line is determined by computing the Orsat analysis for the worst case of imperfect combustion, that is, where none of the active constituents of the fuel burn to CO_2, thus:

$$0.965C\,H_4+1.93O_2+7.3\,N_2 = 0.965CO+1.93\,H_2O+0.482O_2+7.3\,N_2$$
$$0.021\,N_2 = 0.021\,N_2$$
$$0.014CO_2 = 0.014CO_2$$

$$1.0\ \text{gas}+9.23\ \text{air} = 0.014CO_2+0.965CO+1.93\,H_2O+0.482O_2+7.321\,N_2$$

The resulting Orsat analysis will be:

$$\frac{0.014\,CO_2\times 100}{0.014CO_2+0.965CO+0.482O_2+7.321\,N_2}=0.16\ \text{per cent}\ CO_2$$

$$\frac{0.482\,O_2\times 100}{8.782}=5.5\ \text{per cent}\ O_2$$

$$\frac{0.965\,CO\times 100}{8.782}=11\ \text{per cent}\ CO$$

These values, when plotted, determine the lower end of the zero excess air line. Also, a line drawn from the lower right-hand corner through this point determines the sloping base line.

The CO lines are evenly spaced and are drawn parallel to the perfect combustion line. The spacing is such that the 11 per cent line passes through the last point calculated above.

The excess air and deficiency of air lines are parallel to the zero excess air line so that they may be drawn as soon as one point is located on each. A point on the perfect combustion line, for 20 per cent excess air, may be located in the following manner: (Refer to first calculation.)

$$1.0\ \text{gas}+1.2\,(1.93O_2+7.3\,N_2) = 0.979CO_2+1.93\,H_2O+8.781\,N_2+0.386O_2$$

$$\text{therefore}\quad \frac{0.979CO_2\times 100}{0.979CO_2+8.781\,N_2+0.386O_2}=9.75\ \text{per cent}\ CO_2$$

$$\text{and}\quad \frac{0.386O_2\times 100}{10.146}=3.80\ \text{per cent}\ O_2$$

These values of CO_2 and O_2 locate a point on the 20 per cent excess air line and on the perfect combustion line and therefore permit the line to be drawn. The other lines may be located in a similar manner and the chart is then complete.

Atlanta, Ga. NEWTON C. EBAUGH.

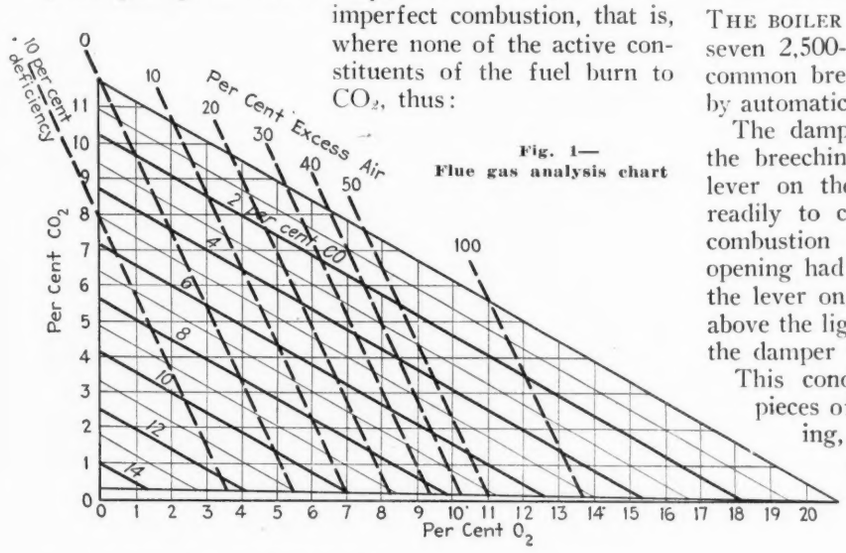


Fig. 1— Flue gas analysis chart

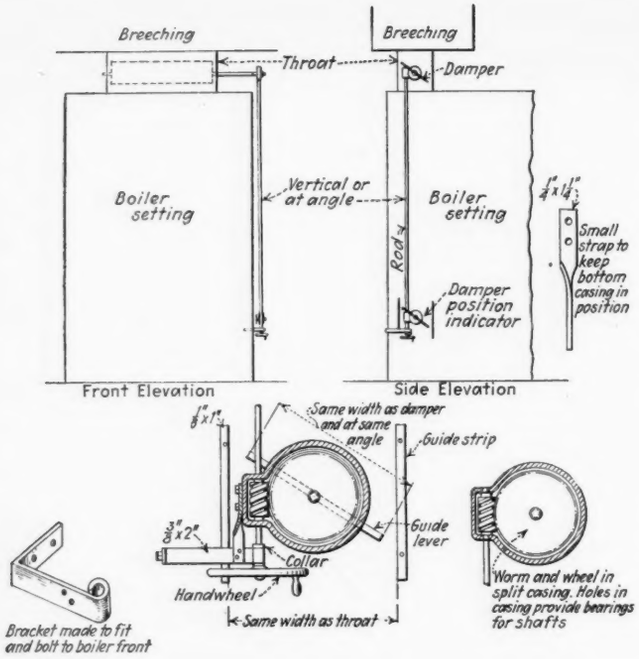
How Close Damper Regulation Was Made Possible

THE BOILER ROOM of a certain plant was equipped with seven 2,500-sq.ft. water-tube boilers all connected to a common breeching. Five of the boilers were operated by automatic stokers and two were oil-fired.

The dampers in the throat between each boiler and the breeching were operated by wires attached to the lever on the damper rods. This did not lend itself readily to correct adjustment of the damper for the combustion conditions existing in each boiler. The opening had to be judged by means of the position of the lever on the rod, and as this was high in the room above the lights it was at times difficult to see how much the damper was open.

This condition was remedied as follows: A few pieces of apparatus, as used in ventilating the building, were obtained and installed to operate the damper. These consisted of handwheels, rods, brackets and sets of worm gearing.

The worm gear and bracket were



attached to the damper rod and the worm on a length of rod which was run down to a convenient height above the floor within easy reach of the fireman. To the end of this rod the handwheel was fastened and a bracket placed just above it and attached to the boiler front to keep the whole set-up rigid.

Above this second bracket another set of worm gearing was placed. This had the same ratio as the set attached to the damper. To the spindle of the worm gear in this case, however, was attached the lever taken from the damper rod. This lever was attached so that it was parallel to the damper, and two strips attached to the boiler setting indicated or represented the throat of the breeching. When the handwheel was turned the motion was transmitted to the damper and to the position-indicating lever simultaneously.

Irvington, N. Y. JAMES R. KNOX.

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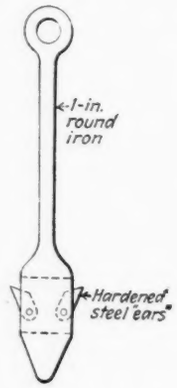
CURRENT COMMENT

Reclaiming Broken Pump Plunger

IN A recent issue of *Power*, F. R. Smith showed a sketch of a tool used for reclaiming the broken plunger of a deepwell pump. It recalls to my mind a similar experience of some years ago.

Here the article to be reclaimed was not a pump rod but 160 ft. of $3\frac{1}{2}$-in. pipe, with an equal amount of $\frac{3}{4}$-in. pipe clamped to the side of the $3\frac{1}{2}$-in. pipe. The $3\frac{1}{2}$-in pipe was the delivery pipe, and the $\frac{3}{4}$-in. pipe was to supply air under pressure for raising water by means of the air lift. The pipe in question was being lowered by means of a chain block into a well 365 feet deep with a 6-in. casing. The chain hoist broke and allowed the pipe to drop and come to rest with its top end 195 ft. from the surface.



"Ears" catch in coupling between two length of pipe

The limited space around the pipe precluded the use of such a device as that shown by Mr. Smink, so a device which would go inside the $3\frac{1}{2}$-in. pipe was constructed as shown in the accompanying sketch. A wire cable was attached to its upper end and a windlass was arranged at the well top for hoisting the pipe.

The tool was lowered and "fished" around until it entered the pipe and dropped below a coupling. Then it was drawn up and the hardened steel "ears" dropped into the space between the ends of two lengths of pipe. The pipe was successfully raised. C. B. HUDSON.

No. Plymouth, Mass.

Condenser Research Shows Effect Of Tube Arrangement

THE article by F. A. Wulfinghoff in the Oct. 13, 1931, number is concerned with the description of tests conducted by Dr. Jungnitz of Berlin Technical University to determine the improvement of the heat transfer coefficient in condensers through reduction of the insulation effect of water dripping on the condenser tubes.

One of the models used in his tests has been called the "Ginabat" arrangement and consequently refers to the condensing system which has been patented under that name. This patent was worked out exclusively by "Société des Condenseurs Delas," Paris (France).

It would appear from the description given that Dr. Jungnitz has not fully understood the "Ginabat" system. He seems to believe that the Ginabat patent refers only to tangential flow of the water; this is in error because it is much more comprehensive. It gives a detailed description of the tube arrangement in two stacks, each of them corresponding to reverse tangential flow. An important feature in the special design of the "Ginabat" arrangement is the air path between the stacks and lanes provided all around for the incoming steam. This special arrangement is not to be found in Dr. Jungnitz's model as it embodies a compact stack for getting only tangential dripping.

The conclusions drawn by Dr. Jungnitz from his tests cannot be held as convincing for the following reasons: The apparatus used for testing has not been built according to the primary rules of condenser construction; in particular the air is extracted by an air pump in six places located on the upper section and on the sides of the condenser, and no care has been taken for cooling this air before aspiration; moreover, the lower part of the testing apparatus remains filled with still air, therefore plays no part in the condensation process.

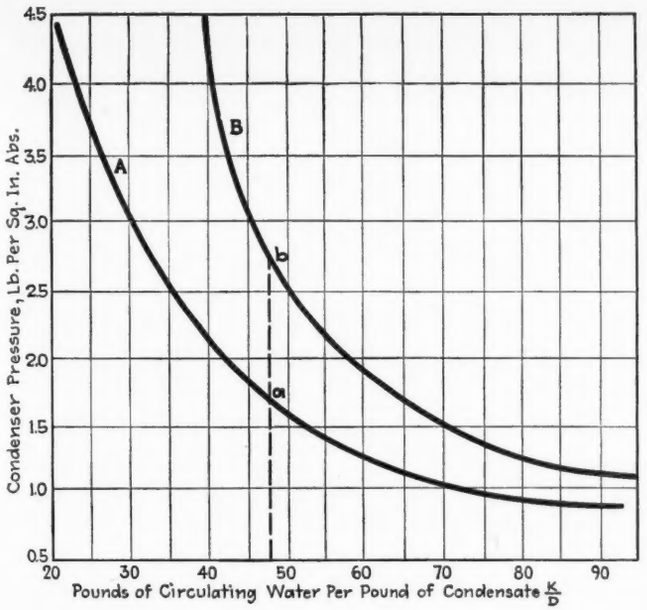
The consequences of these poor arrangements are evident from the results obtained by the experimenter.

1—The great amount of steam aspirated by the ejector while the air in the outlet pipe undergoes a large loss of pressure head.

2—The vacua are quite bad, much worse than those obtained industrially in turbine condensers.

3—The values of the coefficient K determined by the conductor of the tests are, in every case, much smaller than the values given by theory; the comparisons are, in this case, of little value.

Inasmuch as the testing apparatus is partially filled



with still air, the study of the problem of condensate elimination is impossible.

Also, the apparatus is too small to give a good picture of what is going on in a condenser of normal size.

To get an idea of the smallness of the values found for the coefficient K, we will compare them to the values figured out by Professor Ten Bosch in his book: "Die Wärme Uberträgung."

The coefficient K is given by the well known formula (Ser formula):

$$\frac{1}{K}=\frac{1}{\varkappa_1}+\frac{1}{\varkappa_2}+\frac{\delta}{\lambda}$$

Where $\varkappa_1$ is the heat transmission coefficient for steam (its value is 2.100 B.t.u. per sq.ft. per hour and deg. F. — see Ten Bosch — page 214)

δ, thickness of tube.

λ, heat conductivity for brass.

$\frac{\delta}{\lambda}$, is small and may be disregarded.

$\varkappa_2$, heat transmission coefficient for water, is given by following formula: $\varkappa_2=1.140\times fd\times fe\times 0.75w$, where $fd=1.1$ for a diameter of 18 mm. and $fe=0.23$ for a wall temperature of 110 deg. F. and a water temperature of 65 deg. F.

The value of $\varkappa_2$ is thus calculated for the following velocities 5, 10, 15 and 20 ft. per second.

Velocity Feet per Second Values of $\varkappa_2$		B.t.u. per Sq.Ft. per Hour per Deg. F.
5	=	900
10	=	1,610
15	=	2,200
20	=	2,720

Values of K are:	Value of K
5	630
10	935
15	1,030
20	1,180

Dr. Jungnitz has found for the same velocities:

5	$K=175$
10	$K=275$
15	$K=340$
20	$K=400$

The comparison between the foregoing tables shows clearly the inadequacy of the testing apparatus used by Dr. Jungnitz and that the conclusions drawn from the result of the tests are not entirely correct.

If curves from Figs. 2 and 5 are superimposed, it will be noticed that curve A corresponding to the first condenser (Ginabat or ordinary) is continually under curve B corresponding to the plate condenser (Fig. 1).

A curve has been obtained for a constant weight of Steam D_1, the amount K_1 of water being variable, curve B for a constant weight of water K_2, the amount D_2 of steam being variable.

Let us take on curve B point b corresponding to a weight of steam

$$D_2=\frac{D_1\times S_2}{S_1}$$

where:

S_1, surface of first condenser.

S_2, surface of second condenser.

Then point a on curve A corresponds to the same value of the ratio K/Do.

For the points a and b, the condensation rate is the same for the two condensers, the velocity of water being the same in the tubes; but, the vacuum is better in the first test than in the second one.

The logical conclusion of such results would be that the use of plates would reduce the condenser efficiency. The tests conducted by Dr. Jungnitz do not authorize the conclusions that have been drawn from them. It seems to me useless to discuss such an old formula as $K=C\sqrt{W}$.

The coefficient K depends on numerous factors: fluid condition, temperature, surface condition, amount of circulating water, amount of steam, water velocity, steam velocity, etc. . . .

It would be quite dangerous to think that the coefficient K depends only on the ratio of condensed steam to the weight of circulating water.

The coefficient E is equal to Q/θ where Q is the latent heat of condensation for steam and θ the mean difference in temperature of steam and circulating water.

If Q is constant, which is nearly true between given limits, E will be inversely proportional to θ: this is not a new rule for the design of condensers.

With respect to the efficiency of the Ginabat system: the first tests were conducted at the naval arsenal of Indret (France); the results were that the coefficient K was improved in a proportion ranging from 4.5 to 24.6 per cent when the velocity of water in the tubes was increasing from $\frac{1}{2}$ ft. to 3 ft.

The difference in favor of the Ginabat arrangement against the usual one has been emphasized by modern design and manufacture.

The following table compares tests conducted on two identical condensers, one of the two being designed according to the Ginabat system, the other according to the ordinary practice. These two condensers have been erected at the Gennevilliers Power Plant, Union d'Electricité, Paris (France).

	Ginabat	Ordinary
Date of test	Sept. 27, 1923	Oct. 1, 1923
Area, sq.ft.	37,800	37,800
Amount of steam condensed per hour, lb.	262,000	234,000
Absolute pressure at the exhaust inlet, inches (Hg)	1	1.2
Corresponding temperature, deg. F.	79.7	85.2
Inlet water temperature, deg. F.	63.5	64.5
Outlet water temperature, deg. F.	69.4	70.2
Coefficient	534	349

The improvement due to the Ginabat system is a 53 per cent increase in the over-all heat transfer coefficient.

It is not necessary to add that later practice (from the year 1924 to now) has improved the efficiency of the Ginabat system.

Paris, France. ALBERT DELAS.

READERS' PROBLEMS

WILL A NEW ENGINE PAY?—*We are operating an old simple Corliss engine generator in our plant. While its cost was written off many years ago, the engine is still in perfect operating condition. Power is a byproduct for about eight months of the year. Some steam is wasted during the summer months. How can we tell whether the installation of a modern efficient uniflow engine would pay?* J. R. S.

If you will get the following data the answer will be one of fact rather than of opinion: You must find out how much of the steam now wasted yearly would be wasted with the new unit, the value of the steam thus saved and the fixed charges on the new installation. If the steam saved will cover these fixed charges by a safe margin the new unit should be installed.

The difficult problem in your plant, as in many others, is to measure the steam waste. With a pulsating exhaust, a flow meter on the exhaust waste line will not give accurate readings. Still an attempt should be made to measure this waste by an orifice, or otherwise, hour by hour for a typical summer day. For each hour of the same day read the ammeter or wattmeter, and from it (and test data) estimate the total steam consumption of the engine. Steam consumption minus waste for a given hour will give the heating and process load for that hour.

From the electrical loads and the guarantees for the new engine, together with the hourly heating and process loads, figure the waste with it hour by hour.

Thus you can get the waste per day, and for four months, with the present engine and with the new engine and can compute the saving, knowing your steam cost.

TWO-PHASE MOTOR OPERATED ON A THREE-PHASE CIRCUIT.—*I have a 10-hp. two-phase motor that I would like to use on a three-phase line. Is it possible to use the motor for this service without making extensive changes in its windings?* M.R.

If the motor is to be used only temporarily, connecting the two windings to form a "T" may make the motor suitable for three-phase operation. This is done by connecting one terminal of the *A* phase to the middle point of the *B* phase. Then, the remaining *A* terminal and the two *B* terminals are connected to the three-phase line.

This connection does not give a balanced three-phase winding, and if the motor is to be loaded much above 60 per cent of its two-phase rating it may overheat. To obtain a balanced three-phase grouping of the winding it will be necessary to cut out 13 per cent of the coils in the *A* winding. With the same temperature rise as when the motor was operating two-phase the motor will, with the balanced "T" connection, be good for about 87 per cent of the two-phase rating.

If the motor is to be used permanently on three-phase it will be advisable to rewind it for the new conditions. The three-phase winding will have 1.732 times the number of turns in the coils of wire one size smaller than in the two-phase winding. The winding should be connected as star with the same number of circuits in the original grouping.

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Two Practical Babbitting Problems

Readers' answers to two babbitting questions presented in May 10 number give practical suggestions for securing good bond and show how mixtures of old and new metal should be safeguarded

QUESTION 1

FOR babbitting high speed bearings we use alloy No. 2, A.S.T.M. Specification B-23, which is commonly recognized as the original composition of genuine babbitt. New material is purchased in ingot form, but we always have on hand an accumulation of reclaimed ingots made by remelting babbitt chips, which often contain chips of other materials, such as cast iron, steel, brass or bronze. The latter are skimmed off or left in the bottom of the melting pot when casting ingots or reclaimed metal, so that the resulting material is probably of nearly the correct analysis. We should like the opinion of Power *readers regarding the recommended maximum allowable percentage of reclaimed babbitt which may be mixed with new metal for babbitting high-grade, high-speed bearings.* D.L.B.

QUESTION 2

WE ARE experiencing trouble with main bearings in our high-speed Diesel engines, due to poor bonding of babbitt to the cast-steel shells. Will the readers please offer suggestions for improving the bonding by the various methods they have undoubtedly tried? What is an infallible method for detecting poor bonding prior to installing the bearings. Have any tried a scleroscope, or anything similar? Comments will be greatly appreciated. W.A.H.

Remelting Is Good Practice

ANSWERING the question of D.L.B., it is common practice to use all the scrap babbitt on hand, adding enough ingot metal to make up any deficiency. There are good reasons for this. Babbitt, the inventor, used several different formulas. The combination of 85 per cent tin, 10 per cent antimony and 5 per cent copper was his favorite. I believe that most makers of tin babbitts use approximately this formula as well.

At the other extreme is hard lead or type metal, frequently called anti-friction metal. This has its maximum hardness at 83 per cent lead and 17 per cent antimony. (Some makers add tin also, but I think that a small amount of tin added, as in the case of brass, is of negligible value.) These two metals alloy in all proportions, and neither seems to affect the wearing qualities of the other.

In remelting, the iron and steel should be removed with a magnet, as any bright pieces will take up tin and may be carried along. The copper, brass and bronze will not be absorbed to any extent, and will do no harm if they are. The grease is an advantage, as it rises to the surface and protects the metal from oxidation. When the temperature rises high enough, the grease will reduce any oxides of lead, tin and antimony to metal.

In melting babbitt (or solder) the best protective covering I have found is heavy lubricating oil or pine sawdust. An old iron teakettle makes an excellent melting pot, as the babbitt can be poured from the bottom to keep the dross out of the mold.

Much has been said and written about overheating of babbitt while melting, but my observation is that more amateurs pour it too cold than too hot.

FROM W.A.H.'s description of the babbitted bearings on his diesel, I'd say that the manufacturer did a poor job.

If the babbitt used was tin babbitt, it should have been soldered to the frame or auxiliary bearing, then beaten and bored to size. This is done by tinning the inside of the casting, pouring with a mandrel slightly smaller than the shaft, beating and then boring to size. Such a job can hardly get away. If, on the other hand, the babbitt is of the anti-friction, lead-antimony type, pockets should be cut in the frame of the pillow block so deep that the cast babbitt cannot turn.

In 1917 or 1918 a leading maker of anti-friction metal issued a booklet describing the different kinds of bearing metal and their applications. He summed up about as follows: If you are going

to use a thin layer, soldered and hammered into place, use tin babbitt. If not, use the lead-antimony type. Some writers, including Sir John Rankine, say that the tin babbitt is preferable for light loads and high speeds and the other for heavy loads at slow speeds. One firm of my acquaintance has used the latter successfully for 10,000-r.p.m. 24-hour service, replacing bronze successfully.

In the Joplin (Mo.) district, which crushes more and harder rock than any camp in the United States, spelter (common zinc) has been used successfully for bearing metal on crushers and rolls, which get the hardest treatment I know of. I have never known it to slip unless worn very thin, and have seen it worn until the buttons that held it were all that was left. In the early days of shallow diggings, which were here today and gone tomorrow, some of the machinery was fearfully and wonderfully made, but no one has accused it of not standing up under hard service.

St. Louis, Mo. L. R. BAKER.

Favors New Metal

THIS is my answer to D.L.B. Among practical men there is a feeling that any old metal added is sure to cause trouble. This is probably because they have been unable to obtain a clean mixture. The babbitt metals are a mechanical mixture and not chemical, so it is advisable to stir all babbitt thoroughly just before pouring. It should also be skimmed carefully. The heat should not be high enough to turn the metal brownish on top; this increases waste and causes blowholes.

Permissible variations in "genuine babbitt" are given on page 582 of Marks' *Mechanical Engineers Handbook*. From a study of these it would appear that not over 20 per cent of used material should be added.

If bearings are high-speed, care should be taken that none of the harder materials are imbedded in the surface of the bearing. They will cause scoring.

If much delay would be experienced by a bearing failure, or should it be hard to get to or costly to renew, I am sure that the additional cost of new metal would be justified.

BY PROCEEDING as follows, W.A.H. can insure a good bond: The liner should be cleaned with sal ammoniac while it is hot. Then it may be tinned with 50-50 solder. The best way to heat the liner is in a pot of hot metal. A mixture of red clay and water or graphite and water may be rubbed or swabbed on any surfaces which should not be tinned.

Drilling holes at an angle to the bearing surface or drilling holes from $\frac{1}{4}$-in. to $\frac{3}{4}$-in. in diameter straight into the liner will tend to give a bond. It is a good idea to countersink these holes to eliminate sharp corners. Near ends and openings in the bearings it will help to drill a line of small holes. If a shaper is handy, a few shallow longitudinal grooves of dovetailed construction will insure a good bond.

Pouring the shell cold will cause the metal to shrink away from the shell in cooling. Any matter which will produce steam will cause blowholes.

Many recommend peening the metal while it is still hot, claiming that this gives a better bond. I believe that this hammering will loosen the metal from the shell. I have read somewhere of fastening bonds with flat-head machine screws. If the screw is placed in before pouring and is well below bearing surface, I believe that this would be a good method. GRADY H. EMERSON.

Birmingham, Ala.

A Question for Our Readers

OUR HEATING system has just completed its first full season of operation. All of the radiators are equipped with orifices installed at the radiator valve union. As there was considerable pipe scale in the system, a few of these orifices became clogged. Is it advisable to inspect all of the orifices before the next heating season and, if so, what is the best method of cleaning them? H. MCG.

Suitable answers from readers will be paid for if space is available for their publication. Typewritten replies should preferably be double spaced.

▼

Diesel Engine Bearing Manufacturing Practice

PRESENT-DAY manufacturing practice for diesel engine bearings is: Cast steel shells are bored out with a roughing tool. No finishing tool is used. Dovetailed radial slots are machined in the shell; longitudinal slots are also sometimes added. After all machine work is finished the shell is heated to a temperature of 400 to 500 deg. F. to remove all traces of oil or grease, then cooled and put into a 10 per cent hydrofluoric acid pickling solution (sulphuric or muriatic acids are also sometimes used). When the bearing is removed from the pickling solution it is washed in clean running water to remove all traces of the acid. The shell is then tinned (if tinning is to be satisfactory it must be done shortly after removing the shell from acid solution). Zinc chloride solution is swabbed over the surface of the part to be tinned. The part that is not to be tinned should be swabbed with a very thin red clay solution or lightly covered with flake graphite. The shell is then dipped into a pot of molten solder (50-50) and allowed to remain there until the shell reaches the same temperature as the solder (500-600 deg. F.). It is then removed, swabbed with the zinc chloride solution and again dipped in the molten solder. When the shell is again removed it is babbitted immediately, and the tinned surface is not touched.

Some manufacturers centrifugally cast the babbitt metal into the shell. That is they have the shell revolving at a high rate of speed when the babbitt is poured. This is not done to get a better bond between the babbitt metal and the shell. It is to increase the density of the metal, eliminate air pockets and to reduce the rate of wear. Ordinarily however, it is not possible so to cast the bearings in the field. Good bearings can be cast however, providing, a high-grade babbitt is used. It should always be poured at the temperature the manufacturer recommends, and a pyromometer must be used to measure the temperature. The old time-tried method of using a pine stick to determine the temperature has no place in diesel engine work if good long-life bearings are desired.

Although it is against standard practice I strongly recommend cooling the babbitted shell off rapidly. Just as soon as the babbitt is hard carefully remove the shell from the jig without jar, and cool it off by using a sack soaked in water. After ten or fifteen minutes of the latter, plunge the shell into a basin of clean running water.

Machining of the bearing also plays a big part in a good bond between babbitt and shell. A tinned shell should never be peened. If the bond between the shell and babbitt is good it will ring like a bell when struck lightly with a hammer. The higher the pitch of the note produced the better is the bond.

I have not tried a scleroscope and must confess that I can see no possible advantages of such a test. In the first place you would be unable to tell from the rebound of the ball whether you were over a soft spot in the babbitt or whether there was a void between the babbitt and the shell. A. W. STALNAKER.

Osceola, Iowa.

Preheat Shells and Peen the Bearings

IN POURING babbitt bearings, I would suggest that W.A.H. preheat the shells will a blow torch or gas flame to about 150 deg. F. An air hammer and convex tool or a round-face button set hammer should be used to go over the face of the finished bearing. This relieves contractional stresses in the face of the babbitt. He should not assume that babbitting can be done so that the bearing is in effect one solid piece of metal. It can't be done. I have seen hundreds of babbitt bearings and still wait to see one that has not some looseness in it. It is my belief that tinning of marine engine bearings is not only unnecessary but that it actually aids the babbitt in separating, for when the bronze shell heats in operation, unequal expansion forces the babbitt loose at the tinned surface.

Troy, N. Y. JOHN A. NAGLE.

Paints to Protect Against Corrosion

AT THE N.E.L.A. Hydraulic Power Committee's spring meeting, Prof. Colin G. Fink of Columbia University, New York City, gave a lecture on "Paints and Corrosion." This lecture and the discussion on it have been published in a report by the National Electric Light Association, 420 Lexington Ave., New York City, known as Publication No. 222.

Prof. Fink said that the electrochemical theory is generally accepted as explaining corrosion phenomena. There exists on metallic surfaces extremely small differences of potential. As long as the surface is dry these potential differences have no effect on the material. However, as soon as the surface becomes moist, an electrochemical action begins. Certain areas, the anodic ones, go into solution. This change of the surface condition is corrosion. The differences in potential have several causes:

1. Differences in composition—even in so-called chemically pure iron these voltaic couples do exist.

2. If there were no differences in composition from point to point the differences in the orientation of the crystals are sufficient to cause very small potential differences.

3. Physical stress—in a material subject to stress the more highly strained portion acts as an anode. Strains caused by bending or unequal heating are likely sources of corrosion but can usually be eliminated by proper annealing.

4. Difference in surface covering—poor painting may be the beginning instead of the end of corrosion.

Three factors are to be contended with in an attempt to prevent corrosion: water, air and sun. It is a comparatively simple matter to protect metal against any one of the three, but the combination of these forces is very destructive.

Tests Made on Paints

Recent tests have been made on the life and effectiveness of paints by a subcommittee of the Hydraulic Power Committee, N.E.L.A. These tests were of a practical nature and, if properly interpreted, are, in Prof. Fink's opinion, valuable to the industry. Much can be said in regard to the conclusion that red lead increased the life of certain paints. As a result of modern methods of manufacture the red lead is very finely divided and each small particle is surrounded by a cover of oil. When first applied the only protective action of the red lead is mechanical, the lead itself only adds body to the coat of oil. Before the lead becomes a protective influence, the oil must begin to break down and permit the lead to come in intimate contact with the iron. Paints may have been better years ago when the lead particles were so coarse that they could not be completely prevented from touching the metal.

The most effective paint is the one which has the greatest affinity for the material to which it is applied. Preferably the paint should form a compound with the base material. Such a paint is not yet on the market; but we can expect that when its value is realized and a demand is made for its production the manufacturers will be quick to respond. Researches at Columbia University have given most encouraging results.

In the discussion W. P. Creager told of his experience with painted penstocks. Various paints were applied by representatives of paint manufacturers to the inside, outside and buried portions of the pipe. All the paint was gone from the inside of the two penstocks within a year.

Protecting Pipe Line With Bituminous Covering

Prof. Fink described the method of protecting a chemical pipe line exposed to severe conditions. This line, of iron pipe, crossed the New Jersey marshes where it was exposed to sun, air and salt water. Bituminous covering seemed to be the best protection, but the asphalt would melt and run in the high summer temperatures. To prevent this the pipe was wrapped in burlap before applying the bituminous paint. The protective qualities of this covering have been satisfactory. Two or three alternate layers of burlap and asphalt paint are recommended.

Attention was called to how paints lasted longer on a rough or oxidized surface than on a smooth one. It is therefore considered advantageous to allow natural corrosion to roughen the surface and to remove only loose rust by brushing. It may even be advisable to use the cheapest labor which will be the least diligent about cleaning the metal. The paint should be applied over the corroded surface. This applies to all paints. Sand blasting should be used only to remove the surface material from castings and rolled stock. This surface material is brittle and of much poorer quality than the interior metal. After the sand blasting it would be well if rusting were allowed to proceed. It is the practice of several large companies to allow metal structures such as those of corrugated galvanized iron to corrode for a definite period after erection before painting.

It was the opinion of Prof. Fink that if the metal surface is always covered with water there is no need of trying to protect it with paint. After the material has been cleaned submerge it and let nature take its course. The water itself may be more efficient in protecting the metal than paint; in particular does this apply to very hard waters. However, if a metal is removed from acid water and allowed to dry, then in the process of evaporation the acids may become more concentrated and sufficiently active to attack the metal rapidly.

Byron E. White cited experience with trash screens at Trenton Falls. The screens had been submerged under a minimum of 36 ft. of water for 30 years. They were still effective when in 1931 it was desirable to replace them because of the construction of new screens to cover adjacent openings. The thickness of the steel members had been reduced approximately 50 per cent.

Aluminum Paint

The subject of aluminum paint was interjected into the discussion by J. P. Growdon, and Prof. Fink continued: The important thing to remember about aluminum paint is that the proper vehicle must be used. The so-called "Permite-Resalum" paint consists of aluminum-bronze powder in a synthetic resin vehicle. Aluminum bronze is a good reflector of heat and with the proper vehicle aluminum paint may reduce the temperature inside a large gasoline storage tank as much as 10 deg. C. It is standard practice for oil companies to paint their tanks with aluminum and effect large savings in evaporation. From the same line of reasoning we would decide to use black paint for radiating surfaces such as transformers. Aluminum metal is not to be recommended for alkaline conditions. However, it is used for containers of strong acids, such as concentrated nitric acid.

Rubber paints are excellent for use with acid water or moist air, but will not last very long if exposed directly to the sun. Here they require a second outer coating. Rubber paints are tough and wear well.

W. S. Hill described the use of aluminum paint over a bituminous paint. Because of the difference in vehicles which were used the work did not last. However, aluminum-bronze powder can be mixed with bituminous paint to form an excellent but expensive paint. The aluminum will not show and will have no reflecting qualities, but imparts other valuable properties to the bituminous paint.

Although the use of sprayed paint is less efficient as far as amount of paint is concerned, it has, as Prof. Fink showed, one little recognized advantage. The particles of paint hitting the surface at high velocity are more or less squashed. The denser particles penetrate the oil covering and come in direct contact with the metal.

Attention was called to the Schopp Process, which is somewhat similar to paint spraying. Melted metallic lead is sprayed by compressed air onto the steel surface to be covered. Little electrolytic action takes place between the base metal and the lead. Copper, aluminum and zinc can be applied by this process, but tend to form oxides during the spraying.

WHAT'S NEW IN PLANT EQUIPMENT

Low-Head Steam Generator

A RADICALLY new type of low-head bent-tube boiler, to be known as the "S. & G. steam generator," has been perfected by the Freeman Boiler and Engineering Company, 222 W. Adams Street, Chicago. This unit was designed primarily to burn pulverized coal, oil or natural gas, and is suitable for capacities from 15,000 to 100,000 lb. of steam per hour, and for any desired pressure.

Introduction of a bridge-wall across the furnace and modification of the radiant surface forward of the bridge-wall makes the design readily adaptable to stoker firing. The boiler is of the two-drum type, but instead of having the drums across the rear of the setting at right angles to the center line of the furnace, they are placed parallel to the furnace and at one side of it. A double row of tubes completely encircles the furnace, thus greatly increasing the radiant heating surface.

Primary purposes of the design are: to permit greater capacity under limitations imposed by low head room; to provide increased radiant surface without the expense of water walls; to provide an adequate furnace and great length of gas travel.

The furnace design provides a refractory lined pre-furnace to insure ignition. After ignition the gases pass through the radiant furnace section to a large combustion chamber and ash pit at the rear of the setting. The gases then travel forward between and parallel to the drums. It will be noticed that the gas travel is at right angles to the tubes.

Turbulence in the upper drum caused by the rapid circulation through the radiant furnace tubes and by steam liberation is confined to that portion of the drum to the rear of the pre-furnace by a partition across the drum. This partition is cut out below the water line to permit the water level to equalize, and above the water line it is slotted to permit the passage of steam. Back of each slot a deep leg channel is mounted, so that, as the steam passes through this partition, it impinges on the webs of the channels and precipitates the entrained moisture. Steam is then removed from the boiler through a standard dry pipe in the non-turbulent end of the drum.

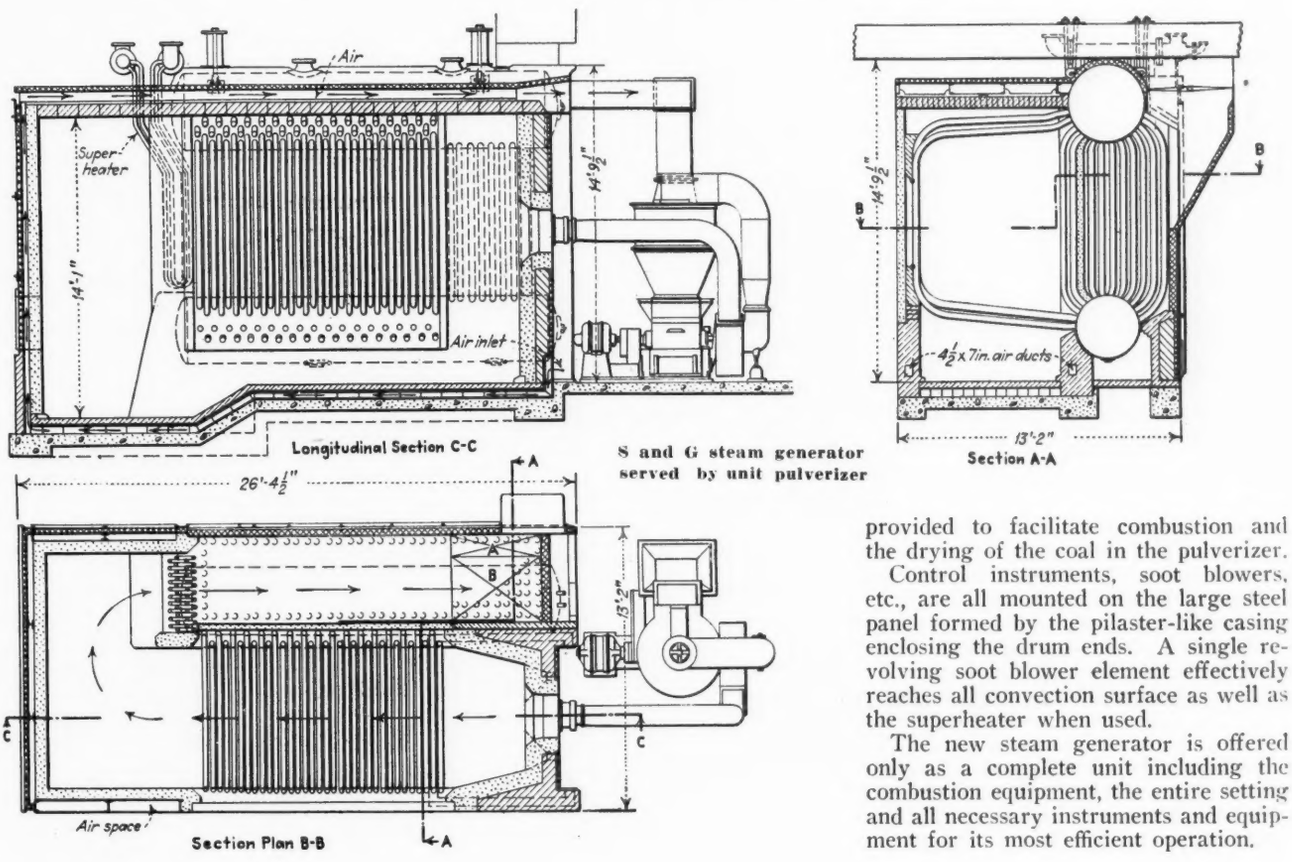
Feed water is admitted through the front head of the drum and by means of a boxlike baffle is forced to pass to the lower drum through the outer two or three rows of tubes indicated at *A* in the drawing. A partition in the lower drum, having a latched door to permit access for cleaning, forces the feed water to return to the upper drum through the tubes indicated at *B*, where it passes under the partition in the upper drum to enter the main circulation of the boiler. The tubes forward of the drum partitions thus form an integral economizer.

The only baffles to gas passage are the longitudinal refractory baffle or partition of a single cross baffle located below the upper drum partition. This latter baffle projects downward from the upper drum, preventing the hot gases from short-circuiting any portion of the convection tubes and insuring that only the coldest gases or those at the bottom of the convection bank will be removed from the setting.

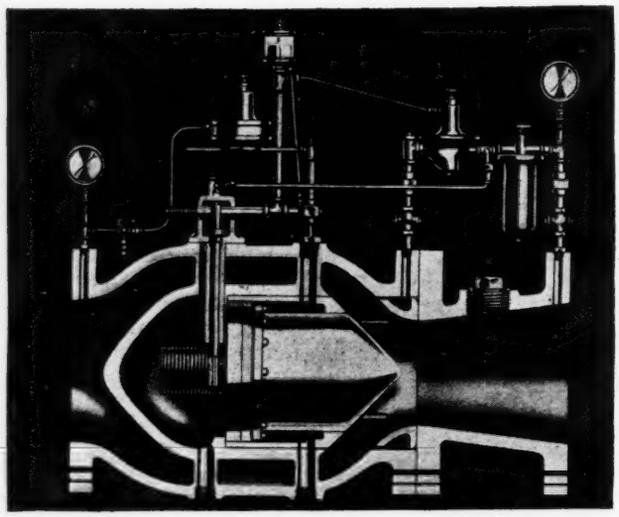
It will be seen that all walls as well as the furnace floor and arch are either air or water cooled, except in the refractory pre-furnace. The life of the refractories is thus protected and a moderate degree of preheated air is provided to facilitate combustion and the drying of the coal in the pulverizer.

Control instruments, soot blowers, etc., are all mounted on the large steel panel formed by the pilaster-like casing enclosing the drum ends. A single revolving soot blower element effectively reaches all convection surface as well as the superheater when used.

The new steam generator is offered only as a complete unit including the combustion equipment, the entire setting and all necessary instruments and equipment for its most efficient operation.



S and G steam generator served by unit pulverizer



Southwark hydraulic valves employing Larner-Johnson principle

Internal-Cylinder Balanced-Plunger Hydraulic Valves

With a view to assisting water works engineers in solving some of the difficult problems met in handling water at high velocity through long pipe lines, and to eliminate dangerous pressure surges and water hammer conditions, the Southwark Division of Baldwin-Southwark Corporation, Philadelphia, Pa., has extended its line of hydraulic valves to include a number of special types.

The Larner-Johnson, internal-cylinder, balanced-plunger principle is employed in the design of many of these valves, which includes slow-closing check valves having an adjustable rate of closure, pressure regulators which function without oscillation or "chugging," surge suppressors which involve an entirely new principle of operation, altitude valves, automatic air valves, throttling valves and automatic gate valves. They are available in all sizes.

Feed Water Regulator

Although developed primarily for service on oil field boilers, the Copes Type DA Regulator put out by the Northern Equipment Company, Erie, Pa., is applicable within certain limitations to other types of boilers. It is a single unit. The valve is installed in a vertical feed line. No brackets or supporting members other than the ¾-in. steam and water connections are needed. The expansion tube is protected throughout its length by heavy channels bolted to the valve body. Movement of the tube acts directly on the valve stem to move the disk, without intermediate leverage. The tube is in tension, a heavy spring at the heel piece end acting as a tension relief. It has a disk type valve and is a check valve as well as a regulating valve. Disk and seat are stainless steel.

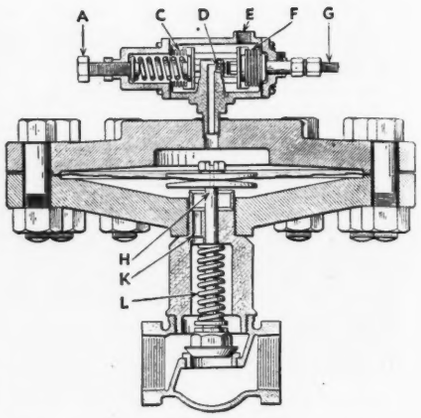
The regulator is furnished only with a 2-in. valve of cast-iron or cast-steel, complete with 250-lb. pressure standard companion flanges.

Copes Type DA feed water regulator

Steam-Operated Controller for Pressure and Temperature

In the steam-operated temperature and pressure controller manufactured by C. J. Tagliabue Mfg. Co., Brooklyn, N. Y., the normal position of the steam valve is closed. Steam enters the valve above the seat and passes at once into the lower diaphragm chamber through opening (K) pushing the diaphragm upward and opening the steam valve. There is also an opening (H) in the valve stem which allows steam to enter the upper diaphragm chamber. If the steam in the upper chamber is not allowed to escape, both chambers will have the same pressure and the valve spring (L) will close the steam valve.

If, however, the steam in the upper diaphragm chamber is allowed to escape through control port (D), which has a larger opening than (H), the pressure above the diaphragm will be less than the pressure below and the steam valve will be opened. The extent of the steam valve opening depends on how fast the control port (D) allows steam to escape from the upper chamber. Hence the steam valve may be wide open, tight shut or in a throttling position.



Tagliabue steam-operated controller

The control port (D) is operated by the thermostatic bellows (F), a part of the usual thermostatic system which includes a bulb and connecting tube (G). (C) is a pressure equalizing bellows and seal for preventing steam from leaking through the pressure and temperature setting screw, (A). (E) is a connection for control port escapement.

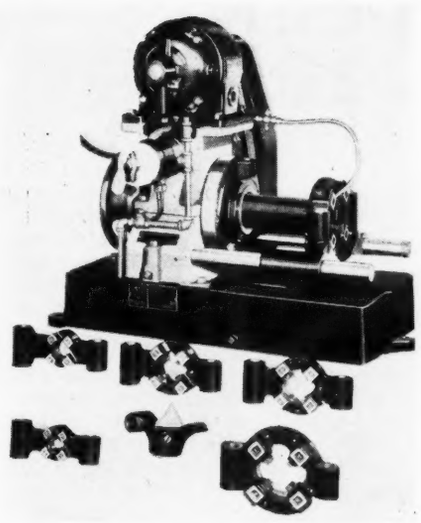
When the bulb is at a lower temperature than that for which the controller is set, the pressure in the bellows is reduced so that it contracts enough to open the control port wide, allowing more steam to escape from the upper diaphragm chamber than can enter through opening (H). The pressure below the diaphragm is now enough to counteract the pressure of the spring (L) and the steam valve opens wide.

Low-Priced Portable Pipe Machine, Capacity ½ to 2 in.

The regular pipe capacity of the No. 512 Tom Thumb pipe machine, put out by Oster Manufacturing Company, 2067 East 61st Place, Cleveland, Ohio, includes all sizes from ½ to 2 in. with an extra capacity down to and including ¼ in. It can be furnished either complete with an oil pump and base containing oil reservoir or arranged for manual oiling.

The machine will also be furnished either with or without a pipe cut-off attachment. The dies are of the segment type held solidly in a holder. The die head is merely slipped on two vertical studs on the carriage. When cutting off, the die head can be swung out of the way to provide clearance for the pipe.

A V-belt is used for driving from the motor to a spiral pinion, then to a spiral gear on the worm shaft. The final drive to the spindle is a triple-thread worm mounted on tapered roller bearings. A three-jawed, quick-acting, universal scroll type chuck is mounted on the



Tom Thumb No. 512 pipe machine

front end of the spindle and a three-jawed universal centering chuck operated by hand wheel is mounted on the rear of the spindle. The motor is universal, variable speed and reversible.

Model GA Water Meter

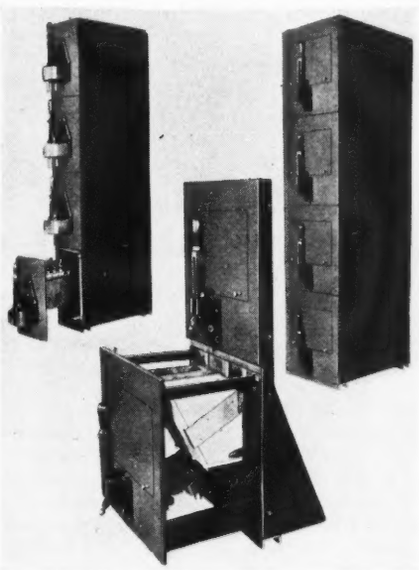
MODEL GA designates this water meter added to the line put out by the Worthington Pump and Machinery Corporation, Harrison, N. J. The meter is built in sizes from $\frac{5}{8}$ to 2 in., with normal flow limits from 1 to 160 gal. per minute. Of the positive displacement disk type, in this meter all register gears are fitted with rubber-bushed bearings. The register box, of heavy construction, is fitted with a double lid to facilitate cleaning of the cover glass. The train assembly comprises four cast bronze gears and four pinions, all rigidly fastened to Monel metal spindles which revolve in replaceable hard-rubber bushings. Although standard construction provides an inclosure for oil, the train may be converted easily to the open type by substitution of an open train plate.

Worthington GA water meter, with flow limits from 1-160 g.p.m.

The disk or measuring chamber is in two sections, held together by a snap joint without screws. Additional protection is secured by the use of a cast-iron frost bottom on the three smaller meters.

Sectional Steel Dead-Front Switchboards and Cubicles

TO MEET the increasing demand for control equipment of the dead-front type, Rollins Smith Company, 233 Broadway, New York, N. Y., has developed the sectional steel dead-front switchboards and cubicles illustrated. The switchboards may be made up of indicating and recording instruments and dead-front knife switches or combinations. Complete switchboards are built up of units each containing one breaker. These units are placed one above the other to form a stack and stacks are placed side by side to form



View of oil breaker stack, air breaker stack and air breaker unit with carriage drawn forward

a section or complete switchboard. Each unit consists of two parts, the frame and the removable breaker carriage. Units for solenoid-operated breakers are constructed in the same manner as the hand operated breakers, except that the handle and target are replaced by a control switch and indicating lamps.

Improved Push Button Switch

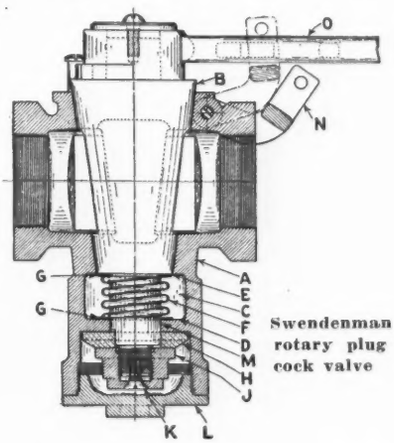
IN THIS Type A-21 push button switch of Lincoln Electric Company, 13034 Coit Road, Cleveland, Ohio, the start button is inside of the stop button. This precludes the possibility of the switch being closed accidentally. The stop button can be operated from any angle by merely striking the front of the control box with the flat of the hand,

Lincoln type A-31 push button switch

but the start button must be operated with the finger. The stop button can be blocked in the off position by pushing it in and twisting it to the right. Heavy springs prevent accidental operation of either button when not locked. The switch is designed for control circuits up to 600 volts a.c. or d.c. Extra-heavy double-break solid copper contacts, which have a positive action, are used. The switch is self-contained and need not be taken apart for wiring.

Rotary Plug Cock Valve

A PLUG cock valve that is said not to stick or leak and does not require lubrication has been developed by J. E. Swendeman, Inc., 171 Camden St., Boston, Mass. The cross-section shows how the tapered plug turns in a tapered seat. The valve casing *A* is provided with a spring chamber *C* with a shoulder or abutment at the outer chamber at *D* and a spring washer at the bottom of plug *B* at *E*. The spring *F* interposed between the abutment *D* and the plug base washer *E* counteracts binding action when turning the plug on its seat. *G* is a lower spring washer. The key collar *H* on the shank prevents adjusting nut *J* from loosening. Safety screw-locking adjustment is at *K*. Other parts are housing cap *L*, plug shank *M*, operating lever locking device *N*, and operating lever *O*.



Swendenman rotary plug cock valve

NEWS of the WEEK

N.E.L.A. Speakers Stress Continued Sound Progress

FEW American industries have been less affected by the depression than that of electric light and power. This fact was noted by several speakers at the Annual Convention of the National Electric Light Association, Atlantic City, June 5 to 10.

Engineering was pictured as a vital process of business by John C. Parker, President of Brooklyn Edison Co. This industry, in particular, said Mr. Parker, stands firmly on the foundation built by engineers. Pointing to the extensive technical development in power station design, Mr. Parker deplored the lack of similar progress in distribution engineering.

In the field of power generation he referred to the extraordinary possibilities in connection with new media for delivering heat energy from boiler to turbines. Much further simplification could be accomplished, Mr. Parker indicated, if fixed ideas and fashions could be set aside in favor of cold-blooded analysis.

Mr. Parker urged the extension of the engineering habit of mind and engineering technique into the non-operative departments. He felt that the accountant can learn as much from the engineer as the engineer can learn from the accountant. In particular, he urged more use of engineer services in determining rates, especially rates on power sold to industry and commercial enterprises.

Another reference to competitive rates was made by T. L. Kennedy, chairman of the Commercial National Section. He said, ". . . nor must we overlook the opportunities to increase sales to commercial and industrial establishments which are quick to take advantage of the use of electricity as a means of improving their operation. Properly to enlarge this market, it is only necessary that we keep our rates at all times competitive with other forms of power and co-operate with industry in the developments of new applications of electricity in production processes."

Several speakers stressed the continuing importance of scientific research in times of depression. Among these were J. S. Tritle, general manager, Westinghouse Electric & Manufacturing Co., and R. A. Neal, central station executive of the same company.

A plea for continued progress was made by Dr. S. M. Kintner, Westinghouse vice-president. "If," he said, "management decides to forget further development for the time being it is on exceedingly dangerous grounds. A modern plant today may be quite out-of-date in a comparatively short time."

A.I.S.E.E. Holds Convention In Pittsburgh Next Week

The convention and Iron and Steel Exposition, under the auspices of the Association of Iron and Steel Electrical Engineers, is scheduled for the William Penn Hotel, Pittsburgh, on June 20, 21, 22 and 23. Included in the papers to be presented are these: "Electrical Developments in the Iron and Steel Industry," by W. H. Burr, electrical superintendent, Lukens Steel Co.; "Safety Regulations Necessary When Working on High-Tension Lines," by the A.I.&S.E.E. Safety Engineering Division (to be discussed by public utilities and steel and industrial plants); "Progress in the Electrical Welding Science," by Prof. G. E. Doan, Lehigh University; "Design and Performance of a Fully Compensated D.C. Machine," by M. E. Thompson, consulting engineer, and I. S. Nippes, electrical engineer, Elliott Co.; "Water-cooled Furnaces for Steam Boilers," by H. J. Kerr and R. M. Hardgrove, Babcock & Wilcox Co.; "Development of Air-Cooled and Insulated Refractory Walls," by R. C. Denny, consulting engineer, and R. D. Foltz, vice-president, M. H. Detrick Co.; "Maintenance and Care of Recording Instruments," by Martin J. Conway, fuel engineer, Lukens Steel Co., and R. C. Kennan, service engineer, Brown Instrument Co.; "Fuel Economy under Curtailed Operating Conditions," by H. Dobrin, consulting combustion engineer, Pittsburgh; "Higher Steam Pressures in the Blast Furnace-Steel Mill Power Plants," by Charles W. E. Clarke, and Walter P. Gavit, United Engineers and Constructors.

Claude Claims to Cure Defects in Sea-Water System

Prof. Georges Claude, who has conducted experiments at Matanzas, Cuba, in extracting power from the sea by use of the warm water at the surface and cold water 3,000 ft. beneath it, believes he has found a way to cure one of the chief defects of the system. Critics had contended that too large a portion of the energy produced by his apparatus was consumed in the extraction of gas contained in the water.

COMING MEETINGS

American Society of Mechanical Engineers—Bigwin Inn, Lake of Bays, Ont., Canada, June 27-July 1. Oil and Gas Power meeting, Pennsylvania State College, June 8-11. C. W. Rice, 33 West 39th St., New York.

American Institute of Electrical Engineers—Cleveland, Ohio, June 20-24. Headquarters, 33 West 39th St., New York.

American Society of Heating and Ventilating Engineers—Hotel Pfister, Milwaukee, Wis., June 27-29. A. V. Hutchinson, 51 Madison Ave., New York.

American Society for Testing Materials—Chalfonte-Haddon Hall, Atlantic City, N. J., June 20-24. C. L. Warwick, 1315 Spruce St., Philadelphia, Pa.

Association of Iron and Steel Electrical Engineers—Annual convention in Pittsburgh, Pa., June 20-24. Managing Director, John F. Kelly, Empire Building, Pittsburgh.

National District Heating Association—William Penn Hotel, Pittsburgh, Pa., June 14-17. D. L. Gaskill, Greenville, Ohio.

University Hunts Data On Life of Corliss

The division of engineering at Brown University, Providence, R. I., has been collecting material concerning the life of George H. Corliss. A great many men served their apprenticeships at the Corliss works and later went out to assume positions of responsibility elsewhere. The university is having trouble to get a record of these men, many of whom should be able to add to the information already available. Among the men connected with the Corliss plant were Alexander Lyman Holley, Nathaniel G. Herreshoff and Edwin Reynolds. The university is attempting to find the names of a number of others.

A.S.M.E. Process Meeting Includes Power Engineering

SEVERAL papers of interest to power engineers were included in the sessions of the Process Meeting of the American Society of Mechanical Engineers, at the Hotel Statler, Buffalo, N. Y., on June 6. Prof. Lincoln T. Work, of Columbia University, in "An Analysis of Crushing and Pulverizing Processes" in the grinding symposium, discussed the different types of crushing and pulverizing mills and showed the diversity in their operating characteristics. Included in his discussion was some information on pulverizing coal. Differences between anthracite and bituminous are sufficient to create a serious problem in the selection of a mill or in the estimation of its capacity. Differences are so pronounced that a mill which will adequately meet the demands with bituminous coal may not even produce a combustible powder when grinding anthracite. Also, the capacity of the mill may be reduced more than half by a change from bituminous to anthracite. Noticeable differences in mill capacity and in power consumed per ton of product are to be observed with different samples of bituminous.

Included in the papers at the heat transmission session were "The Flow of Gases in Liquids and Conduits," by R. J. S. Pigott, staff engineer, Gulf Refining Company; "Design Data and Alignment Charts," by T. H. Chilton, A. F. Colburn, R. P. Genereaux, and H. C. Vernon, of the Experimental Station, E. I. duPont de Nemours & Company; and "Fluid Friction at Parallel and Right Angles to Tubes and Tube Bundles," by E. N. Seider and N. A. Scott, Jr., Foster Wheeler Corporation.

Mr. Pigott's paper gave the results of studies on the flow of liquids and gases in closed conduits, varying from smooth brass pipe to brick ducts. The author has collected all the available authoritative data, particularly with a view to settling the effect of roughness of the conduit. His conclusions show that friction factors for all types of rough surfaces approach the values for smooth conduit as diameter increases. The resistance of very small brass tube; 6 to 12-in. steel pipe; 9 to 16-in. galvanized; 20 to 48-in. best cast iron:

smooth cement; and light pressure plate ducts; 48 to 96-in. average cast iron and average concrete; 72 to 120-in. brick tunnels, is the same. A similar diameter relation holds throughout the ranges of size.

The discussion of Mr. Pigott's paper brought out the existence of much disagreement on the proper corrections for roughness in conduits. Mr. Pigott explained that he is now developing formulas by which roughness may be allowed for without the necessity of using cut-and-try methods.

Mr. Pigott brought out that, contrary to general opinion, the pressure drop due to a bend does not necessarily decrease as the radius of the bend is increased. Beyond a certain radius the additional pipe length added by the larger radius introduces a loss that offsets the gain due to more gradual curvature.

The design alignment charts have been developed by duPont for use in fluid flow and heat transfer problems and include charts on pressure-drop in clean steel pipe; turbulent region; film coefficient of heat transfer, gases flowing through tubes; design chart, gases in tubes; film coefficient of heat transfer, gases flowing across tube banks; free convection outside horizontal cylinders; free convection, vertical surfaces; and film coefficient of heat transfer, condensing vapors. The paper by Messrs. Seider and Scott describes tests and a discussion of results with data, curves, etc., covering fluid friction for heating, cooling and for iso-thermal of flow.

Dust Problems Session

One session of the meeting was devoted to dust problems, with particular reference to the effect of dust on health.

Dr. A. E. Russell, senior surgeon, U. S. Public Health Service, exhibited the detrimental effects of silicosis on the lungs of persons exposed to dust. "It has been established beyond question," said Dr. Russell, "that dust in the lungs promotes pneumonia and other pulmonary troubles." Dr. Russell, as well as J. J. Bloomfield, emphasized the fact that the composition of dust is as important as its quantity in determining the dangerous effects of the nuisance. Mr. Bloomfield expressed the belief that the particle count is at present the best index of atmospheric pollution.

Electric Plant at Arietta, N. Y.

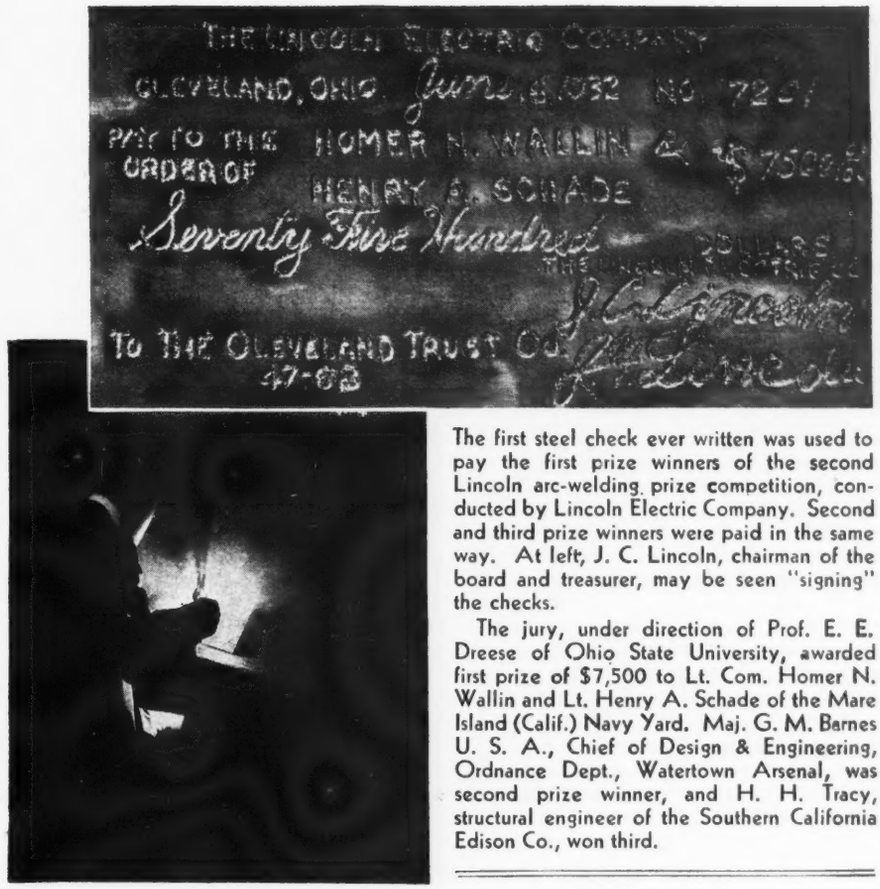
A petition of the New York Power & Light Corporation for authority to construct an electric plant and exercise a franchise in the town of Arietta, Hamilton County, New York, has been granted by the Public Service Commission. The town of Arietta is one of the smallest towns in the State but the population is considerably increased during the summer months. The company proposes to extend its present distribution system from Lake Pleasant.

Puget Sound P. & L. Accepts Equipment

Puget Sound Power & Light Co. has formally accepted the equipment installed in two completed, 20,000-hp. units of the Rock Island hydroelectric project on the Columbia River, according to W. H. McGrath, vice-president. Two of the four units are in operation and performing better than specified requirements, he said. The acceptance covers generators by General Electric Co., transformers and switches by Westinghouse Electric & Manufacturing Co., and water-wheels by Allis-Chalmers. The other two units, each of 20,000 hp., are being installed together with electrical equipment, and will be completed and tested by the end of this year, the time scheduled. About 100 men are now engaged on the project. Later in the year, when western Washington streams are low, the Rock Island plant will be able to furnish all power necessary without recourse to steam generation, according to Mr. McGrath. Construction equipment at Rock Island is being sold and the townsite cleared of all buildings, all major work having been completed.

Change Boiler Inspection System In District of Columbia

The Committee on the District of Columbia, to whom was referred H.R. 8013 to amend the act to "Create the Office of Steam-Boiler Inspector for the District" has reported favorably upon it. The report has been printed as No. 1384, under date of May 19. The bill proposes to abolish the existing fee office of boiler inspector, the last remaining fee office in the District government, and to authorize the commissioners to appoint two or more persons, one as chief inspector of steam boilers, and one or more as assistant inspectors of steam boilers, who shall be paid on an annual basis. The existing fee for the inspection of a steam boiler, $5, will be superseded by a graduated fee.



The first steel check ever written was used to pay the first prize winners of the second Lincoln arc-welding prize competition, conducted by Lincoln Electric Company. Second and third prize winners were paid in the same way. At left, J. C. Lincoln, chairman of the board and treasurer, may be seen "signing" the checks.

The jury, under direction of Prof. E. E. Dreese of Ohio State University, awarded first prize of $7,500 to Lt. Com. Homer N. Wallin and Lt. Henry A. Schade of the Mare Island (Calif.) Navy Yard. Maj. G. M. Barnes U. S. A., Chief of Design & Engineering, Ordnance Dept., Watertown Arsenal, was second prize winner, and H. H. Tracy, structural engineer of the Southern California Edison Co., won third.

PERSONALS

Samuel Insull, veteran public utility magnate, has retired as chairman of Commonwealth Edison Company, People's Gas Light & Coke Company and Public Service Company of Northern Illinois. James Simpson, chairman of the board of Marshall Field & Co., was elected to succeed Mr. Insull as head of the three companies.

Dr. Ambrose Swasey of Cleveland, past-president of the A.S.M.E., and Dr. Philip Lenard, of Heidelberg University, have been awarded Franklin medals and certificates of honorary membership at this year's meeting of the Franklin Institute. Philip Thornton Dahiell, United Gas Improvement Co., Philadelphia, Pa., received the Clark Medal at the same meeting.

Marcy L. Sperry has been elected president of Washington Gas Light Co., Washington, D. C. Mr. Sperry goes to Washington from Fall River Gas Works Co., Fall River, Mass., where he was vice-president in charge of operations.

Luther D. Burlingame, 67, organizer and long a president of Providence (R. I.) Engineering Society, and industrial superintendent of Brown & Sharpe Mfg. Co., Providence, died there June 2.

John A. Drew, 80, for sixty years connected with Worthington Pump & Machinery Co., Harrison, N. J., and at one time general manager, died May 25 in Brooklyn, N. Y.

Richmond Talbot, 59, member of the engineering firm of Sanderson & Porter, New York City, died at his home in Tuxedo Park, N. Y., May 26.

How's Business?

The critical and highly political month of June starts with most business indicators slowly sagging a bit early into the usual slackness of the summer season. The only conspicuous exception is the strengthening of building contracts and collateral structural steel awards, supported principally by public building, which apparently has not yet been completely abolished by public opposition and the prevailing economy complex. Commodity prices continue to weaken. Response of security markets to promise of private banking pool support was sharp but short-lived. Congressional uncertainties and anxiety about European developments still overshadow business and financial sentiment, but one by one the alibis are being overtaken by events, and we are being brought face to face with the fundamental issues of recovery which now lie in the field of monetary policy and must be met by international action. For the moment the most encouraging feature is the slow strengthening of the banking position, with accumulating evidence that the more destructive aspects of credit deflation are temporarily under control and powerful forces toward expansion are beginning to operate. Electric power output, according to the N.E.L.A., was 1,381,452,000 kw.-hr. for the week ending June 4—*The Business Week*, June 15.

BUSINESS NOTES

FISHER GOVERNOR Co., Marshalltown, Iowa, has established two exclusive representatives, the Grinnell Co., 240 Seventh Ave., South, Minneapolis, Minn., and the Steam Plant Equipment Co., 1012 North Third St., Milwaukee, Wis.

H. V. SCHIEFER, formerly with R. H. Beaumont Co., is now associated with C. O. Bartlett & Snow Co., 6200 Harvard Ave., Cleveland, Ohio, in connection with the sale of fuel and ash handling equipment and bulk material handling systems.

L. MUNDET & SON, INC., Commerce & Palmer Sts., Houston, Tex., has established a sub-branch office at 4600 Gaston Ave., Dallas, Tex., in charge of A. A. Stone. Walter R. Rochow replaces Mr. Stone at the Houston office and will assist Willard Selle, manager at Houston.

AMERICAN CABLE Co., New York City, has licensed Wickwire-Spencer Steel Corp. to manufacture preformed wire rope under American Cable patents.

DENVER FIRE CLAY Co., P. O. Box 1107, Denver, Colo., has appointed S. Herbert Lanyon, 408 Call Bldg., San Francisco, an exclusive northern California representative in the sale of industrial gas burners.

New Bulletins

LUBRICATION—Technical bulletin on colloidal-graphited lubricants, Acheson Oildag Company, Port Huron, Mich.

BALL BEARINGS—"Auburn Ball Bearing Data Book," Auburn Ball Bearing Company, 57-85 Clarissa St., Rochester, N. Y.

WELDING—Section 3304, supplies for arc welding, Lincoln Electric Company, Cleveland, Ohio.

STOKERS—Bulletin on "Building Combustioneer." Combustioneer, Inc., Columbia St. and Bechtle Ave., Springfield, Ohio.

SWITCHES, MOTORS, COMPRESSORS—GEA-1568 on magnetic switches, GEA-1542 on Type B direct-current motors, and GEA-1344A on single-stage centrifugal compressors, General Electric Company, Schenectady, N. Y.

GENERATORS—Bulletin 55, diverter-pole generators for motive power batteries; Electric Products Company, Cleveland, Ohio.

SMOKE ABATEMENT—"Black Wings of Smoke," data on smoke prevention activities, Anthracite Institute, 90 West St., New York, N. Y.

BELTING—Booklet on "Garlock Transmission Belting," Garlock Packing Company, Palmyra, N. Y.

TEST OF DAM MODELS—University of Iowa, Iowa City, Iowa—Bulletin 2, New Series No. 230, dated July 1, "Laboratory Tests on Hydraulic Models of the Hastings Dam."

New Plant Construction

Compiled by the McGraw-Hill Business News Department, Which Is Prepared to Furnish a More Complete Daily Service to Those Who Wish It

Calif., Vernon — City Council, awarded contract for construction of a 170 x 240 ft. power plant on Soto St., to Mittry Bros. Constr. Co., 5531 Downey Road, Los Angeles, $206,525. H. S. McCurdy, City Hall, is city engineer.

Calif., Victorville—City, voted $50,000 bond issue for purchase of water system of Appleton Water Co., and plans extensions and additions, including pumping plant, reservoir, etc. C. L. Foulke, 455 4th St., San Bernardino, is engineer.

Colo., Palisade — Bureau of Reclamation, Denver, awarded contract for construction of surface buildings to house $215,000 hydro-electric plant, to H. Shores, Grand Junction.

D. C., Wash.—Dept. Public Buildings and Parks, will receive bids until June 21 for construction of a central heating plant with distribution system, to various buildings, to be equipped with five 1,000 hp. boilers, and all necessary pumps, fans, oil tanks and other equipment.

D. C., Wash.—Treas. Dept., at office Superv. Archt., will receive bids until June 23 for construction of complete steam distribution system, incl. tunnels, conduits, etc., extending from central heating plant to various public buildings.

Fla., Pensacola — Bureau Yards and Docks, Navy Dept., Wash., D. C., taking bids for an air conditioning system, together with underground water mains and electric manhole and electric ductions at naval air station.

Ill., Greenville — Bd. Local Improvements, awarded contract for construction of pump plant, including duplex sewage ejectors, sewage basin and frame building in connection with General Sewer Dist. 2, to P. F. Weilepp & Co., Decatur, $13,271.

Ind., Columbus—City, awarded general contract for construction of a municipal light plant to H. Vogt, Louisville, Ky.; stokers to Westinghouse Electric Co., 621 Guaranty Bldg., Indianapolis. Estimated cost is $40,000.

Ind., Greencastle — De Pauw University, awarded general contract for construction of a heating plant to Henry Vogt Co., Louisville, Ky. Estimated cost is $40,000. Bevington & Williams, Pythian Bldg., Indianapolis, are engineers.

Ind., Hammond — Lever Bros., Hammond, awarded contract for construction 4 story, basement, addition to soap factory and power plant improvements, to F. C. Rowley, Hohman St. Estimated cost is $200,000.

Ia., Sumner—City, will take bids in July for construction of a power and light plant, including equipment. Estimated cost is $95,000. H. L. Cory Company, 1034 Redick Tower Bldg., Omaha, Neb., is engineer.

Kan., Wichita — U. S. Veterans' Bureau, Washington, D. C., awarded contract for construction of U. S. Veterans' home, including refrigerating and ice making plant, etc., to H. D. Ryan Co., 500 North Dearborn St., Chicago, Ill., $513,000; plumbing, heating and electrical work to Connor & Ripstra, 1015 East Douglas Ave., $180,616.

Mass., Scituate — Water Department plans construction of concrete pumping station. Estimated cost is $10,000. Metcalfe & Eddy, Statler Bldg., Boston, are engineers.

Mass., Quincy — Metropolitan Dist. Comn., State House, Boston, plans construction of a sewage pumping station at Houghs Neck serving Germantown, Weymouth and Braintree, also sewer lines, funds appropriated. Estimated cost is $100,000 or more. Metropolitan Dist. Comn., engineers in charge.

Minn., Elk River—City, plans construction of municipal electric light and power plant. Estimated cost is $85,000.

Mo., Illmo—City awarded contract for waterworks improvements including well, pumping equipment, tank on tower, distribution mains, etc., to John J. Miller, Illmo. $37,000.

Mo., St. Louis—U. S. Treas. Dept. Supervising Architect's office, Wash., D. C., taking bids until June 24 for alterations to boiler plant.

N. H., Durham—University of New Hampshire, E. H. M. Lewis, Durham, awarded contract for construction of tunnel for heating system extension to Davison & Swanberg, 61 Amherst St., Manchester, $6,456; heating contract to Jennison Co., Main St., Fitchburg, Mass., $28,500.

N. J., Hoboken—H. Deile & Co., 84 Jackson St., taking bids general contract for alterations to refrigeration plant at 84 Jackson St. Estimated cost is $40,000. Const. Service Corp., 80 Broad St., Elizabeth, is engineer.

New York—Northern New York Utilities Co., Inc., Light & Power Bldg., Watertown, applied for permission to construct electric plant and distribution system in Alexandria and Watertown.

N. Y., Batavia — U. S. Veterans' Bureau, Wash., D. C., will receive bids until July 12 for construction of general medical and surgical hospital, separate contracts for heating, electrical work, refrigeration and ice plant, water softening system, etc.

N. Y., Dannemora—Department of Correction, State Office Building, Albany, awarded contract for construction of steam duct and tunnel from Clinton Prison power house, to Boyojohn & Barr, Warwick, $27,774.

N. Y., Pitcairn—Harrisville Electric Light & Power Co., Harrisville, applied to Public Service Comn., for permit to construct an electric plant and distribution lines in Town of Pitcairn and part of St. Lawrence County.

O., Bowling Green—State Dept. P. Wks., T. S. Brindle, Columbus, will take bids until June 17 for the construction of a power house addition, and equipment, incl. water softener, new stack, etc., at State College.

Va., Quantico—Spec. 6809—Bureau of Yards and Docks, Navy Dept., Wash., D. C., taking bids for extensions of steam mains to quarters and replacement of coal burning boilers and water heaters with steam converters and coil type water heaters at Marine barracks.

Man., Brandon—Manitoba Power Commission, 146 Notre Dame Ave., E., Winnipeg, plans extension of steam heating on Lorne Ave. West. A. Trott, 146 Notre Dame Ave. E., Winnipeg, is engineer.

Equipment Wanted

Engines—Washington, D. C.—Quartermaster, Marine Corps., Wash., D. C., will receive bids until June 22 for 12 field gasoline engines. Sch. 941.

Motor Generator Sets—Wash., D. C.—Bureau of Supplies and Accounts, Navy Dpt., postponed taking bids from June 7 to June 21 for motor generator sets, Sch. 8131.

Pumps—Illinois—War Dept., U. S. Engineer Office, 333 North Michigan Ave., Room 1325, Chicago, will receive bids until June 22 for furnishing and installing 3 vertical irrigation pumps and construction of the Goose Lake Pump house in Grundy County.

Boiler and Condenser—Springfield, Ill.—City taking bids until June 27 for addition to power and light plant including condenser and boiler. Burns & McDonnell Eng. Co., 406 Interstate Bldg., Kansas City, Mo., are engineers.

Pump, etc. — Galva, Kansas — City having plans prepared for construction of a complete waterworks system, including well, pump, etc. Estimated cost $15,000. F. E. Devlin, Wichita, is engineer.

Stoker, etc.—Belchertown, Mass.—Commonwealth of Massachusetts, Dept. Mental Diseases, Boston, taking bids until June 21 for boiler plant alterations, incl. stoker, smoke flue, water tube boiler, etc. Estimated cost is $25,000.

Power Unit — Brooklyn, N. Y. — Spec. 71-4660A, Annex 3—Signal Supply Office, Army Base, 58th St. and 1st Ave., will receive bids until June 20 for power unit, s.c. type PE-*40, per U. S. Army tentative specifications.

Booster Pumps—Youngstown, O.—City, D. Parish, Water Comr., authorized Water Commission to receive bids until June 18, for furnishing and installing 4 booster pumps for city water system. G. Turner, City Hall, is city engineer.

Slide Valve Engine—Trenton, Ont.—Trenton Cooperage Mills Ltd., in the market for one 12 x 14 slide valve engine.